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U. S. S. *NEW MEXICO.*

DESCRIPTION AND OFFICIAL TRIALS.

By Commander S. M. Robinson, U. S. N., Member, and Henderson B. Gregory, Associate.

The electrically-propelled battleship, U. S. S. *New Mexico* is one of three vessels authorized by an Act of Congress approved June 30, 1914. She is officially known as Battleship No. 40, and is a sister ship of the U. S. S. *Mississippi* and U. S. S. *Idaho,* which are equipped with direct-drive Curtis and Parsons turbines, respectively, and geared cruising turbines.

The contract for the electrical propelling machinery for the *New Mexico* was awarded to the General Electric Co., June 3, 1915, at a cost of $431,000.00, and four months later, October 14, 1915, the keel of the vessel was laid at the Navy Yard, Brooklyn.

The *New Mexico* was launched April 23, 1917, and was commissioned on May 20, 1918, since which date she has been subjected to numerous and extensive tests, as noted elsewhere.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	600–0
on L.W.L., feet and inches	600–0
over all, feet and inches	624–0
Breadth, extreme, on L.W.L., feet and inches	97–4½
molded, feet and inches	97–0
Depth molded, main deck at side M.S., feet and inches	46–3
Draught, mean, to L.W.L., feet and inches	30–0
Displacement corresponding, tons	32,000
per inch at L.W.L., tons	100.9
Ratio of length to beam	6.16
Coefficient of fineness, block	0.635
midship section	0.981
L.W.L. plane	0.726

GENERAL DESCRIPTION OF HULL.

In profile the *New Mexico* closely resembles the *Pennsylvania* class. She has one smoke pipe, four turrets and two masts of the customary cage type, which are provided with spotters' tops, wireless, signal yards, etc. There are eight 36-inch searchlights, four on each mast, and four 24-inch signal searchlights mounted on the signal bridge.

Superstructure Deck and Bridges.—The superstructure deck extends from turret II to frame No. 87. On it are mounted four 5-inch guns, the anti-aircraft guns, saluting guns, stowage of boats, foundry and blacksmith shop, deck radio station, etc. Above the superstructure deck is the bridge deck, where the chart house is located. The navigating bridge is on top of the chart house and just behind the conning tower.

Upper Deck.—This deck extends from the bow aft about half the length of the vessel. Forward are the windlass, an electric deck winch and turrets I and II. Aft of the turrets a portion of the 5-inch battery is mounted in a casemate, which also contains the officers' and crew's galleys, butcher shop, bakery, etc.

Main Deck.—The main deck is a weather deck abaft the break in upper deck. In the covered portion forward are

crew's quarters, and aft on the weather portion are turrets III and IV, two electric deck winches and an electric capstan.

Second Deck.—From forward aft are located chief petty officers' quarters, sick bay, crew's space, junior and warrant officers' quarters, wardroom officers' quarters, and the Captain's and Admiral's quarters.

Third Deck.—On this deck, beginning forward, are stores, windlass machinery, prison, firemen's washrooms, coppersmith's outfit, boat crane machinery, radio room, cold-storage rooms, laundry, general workshop, engineer's tool issuing room, crew's space, provisions, and wardroom staterooms aft.

The first and second platforms and the hold are next below in the order named.

COMPLEMENT.

The ship's normal complement, not including additional personnel for flagship, is approximately as follows:

Commanding officer	1
Wardroom officers	26
Junior officers	18
Warrant officers	12
Crew, including chief petty officers	1,011
Marines	72
Total	1,140

BATTERY.

The main battery comprises twelve 14-inch guns, arranged in four three-gun turrets along the center line of the vessel. Turrets I and II are grouped forward on the upper deck, the latter firing over the top of the former, and IV and III are similarly arranged on the main deck aft.

The secondary battery, of fourteen 5-inch rapid-fire guns for torpedo defense, is divided starboard and port as follows: four on superstructure deck and ten within inclosed portion of upper deck.

The 5-inch battery is served by eighteen electric-chain ammunition hoists, each driven by a 3-horsepower motor.

The following smaller guns are also provided:

4 3-inch anti-aircraft guns;
4 6-pounder guns for saluting;
2 1-pounder guns for boats;
2 3-inch field pieces;
2 0.30-caliber machine guns.

The torpedo equipment consists of four 6.8-meter by 21-inch submerged torpedo tubes.

SMALL BOATS CARRIED.

The following small boats constitute the ship's regular allowance:

2 50-foot steamers;
2 35-foot motor boats;
2 50-foot motor sailing launches;
2 40-foot motor sailing launches;
1 40-foot Admiral's barge;
1 31-foot racing cutter;
2 30-foot whale boats;
2 20-foot dinghies;
2 14-foot punts.

All boats are carried on the superstructure deck, port and starboard, except the whale boats, which are hung in davits at the ship's side abreast the after cage mast.

Two electrically-operated boat cranes are provided for handling all boats, except the whale boats. Each crane has two operating gears, one for turning and the other for hoisting, each driven by a 50-horsepower motor.

ANCHOR WINDLASS.

An electric windlass, designed by the American Engineering Co., is located on the upper deck forward, with driving gear below on the first platform. The windlass is driven by

two 125-horsepower motors, through worm gearing and shafting, and so arranged that the wildcats can be operated together or independently of each other.

DECK WINCHES.

Three electric-driven, compound-geared deck winches are provided, one forward on the upper deck, abaft of the windlass, and two aft of the main deck, port and starboard, forward of turret III. They are of the American Engineering Co.'s type, each driven by a 45-horsepower motor.

CAPSTAN.

An American Engineering Co.'s electric capstan is located on the main deck, center line, abaft turret IV, with driving motor below on the third deck. It is compound geared and driven by a 45-horsepower motor.

STEERING GEAR.

The steering gear is of the Waterbury Tool Company's electro-hydraulic type, operated by 75-horsepower motor, fitted in duplicate.

LAUNDRY.

The laundry is located on the third deck, starboard side, midship. It is equipped with the following machinery, all motive power being electricity:

2 Universal presses;
1 Combination ironer;
1 Flatwork ironer;
1 Collar shaper;
2 Extractors;
1 Drying room, 5 drawers;
1 Tumbler dryer;
1 Disinfector;
1 Washer;

1 Ironing table;
1 Soap tank;
1 Starch kettle;
1 2-compartment tub;
3 Truck tubs.

GALLEY OUTFIT.

Officers' Galley.—The officers' galley is equipped with oil-burning range of four sections.

Crew's Galley.—In the crew's galley are installed the following:

1 oil-burning range of eight sections;
4 steam-jacketed copper kettles (80 gallons each);
2 steam-jacketed copper kettles (60 gallons each);
2 coffee urns, 100 gallons each;
1 electrically-driven meat grinder;
1 electrically-driven kitchen and cake machine;
1 hand operated meat slicer;

Bakery.—The following equipment is installed in the bakery:

2 No. 1 U. S. Navy standard, electric bake ovens;
1 steam box;
2 portable dough troughs;
1 electrically-driven dough-mixing machine;
1 electrically-driven kitchen and cake machine;

Miscellaneous equipment located as noted:

1 electrically-driven potato peeler, in potato-peeler room;
1 electrically-driven dish washer, capacity 6,000 pieces per hour, in general mess pantry;
4 warming ovens, in serving room;
1 electric butter slicer, in general mess issuing room.

GARBAGE PLANT.

For disposal of garbage, one oil-burning incinerator is located on the superstructure deck, abreast of the smoke pipe, starboard side.

DRAINAGE SYSTEM.

The four fire and bilge pumps in the center engine room have 5-inch suctions from the after drainage system, which serves compartments aft of number three fire room.

The main circulating pumps, also located in the center engine room, are arranged for pumping on the engine-room bilges in emergencies. There are two bilge connections for the center engine room, and one for each wing engine compartment; each connection is 21 inches diameter.

Each fire room is normally drained by the fire and bilge pump located therein. For extreme emergencies two electrically-driven, vertical, centrifugal pumps are provided for each fire room, with driving motors, 39-horsepower each, located above on the third deck. These pumps have 12-inch suctions and 10-inch discharges.

The drainage of compartments forward, except the forward dynamo room and double-bottoms beneath, is handled by an independent system, served by a horizontal, motor-driven, centrifugal pump. The driving motor is 22.3-horsepower. The latter are drained by the fire and bilge pump in fire room No. 1.

FIRE MAIN.

The fire main is supplied by seven fire and bilge pumps, located in the center engine room and fire rooms.

The main is entirely below the third deck and extends throughout the machinery spaces, close under the deck, in two 8-inch lines, port and starboard. It is cross-connected at the forward and after ends, and in the engine compartments, and supplied by 5-inch risers from the fire and bilge pumps, with cutout valves at the main. From the forward and after cross-connections, 5-inch branches extend in single line forward and aft, reducing in size at the ends 4 inches. Branches to fire plugs are provided as required.

On the main deck forward is a connection to the independent sanitary system for the crew's water closets and washroom. There is a stop valve where it joins the latter.

SANITARY SYSTEM.

The sanitary main is 5 inches in diameter, and is supplied by five 5-inch risers from the fire and bilge pumps, a 5-inch connection from the fire main in the center engine room, and two 5-inch by-passes, one forward and one aft, from the fire main.

Branches, as required, are led to the chief petty officers' washroom and water closets, sick bay, bath, laundry, general mess pantry, galleys, bakery, firemen's washrooms, junior, warrant and wardroom officers' lavatories and water closets, etc.

Forward there is an independent system for the exclusive use of the crew's washroom and water closets. This system is supplied by two 9.3-horsepower, motor-driven, direct-connected, centrifugal pumps, of about 500 gallons per minute capacity each. The main is 5 inches in diameter, with the necessary branches to the plumbing fixtures.

FRESH WATER SYSTEM.

The following fresh-water tanks are provided:

	Gallons.
Ship's tanks	57,910
Mess attendants' washroom supply tank	100
Chief petty officers' washroom supply tank	150
Firemen's washroom supply tank	500
Battle-dressing station supply tank	250
Laundry tank	200
Crew's washroom tanks	450

The ship's tanks have 2½-inch filling connections from the ship's sides, and also a 2½-inch connection from the distiller main. From these tanks water is pumped through the fresh-

water main by means of three direct-connected electric centrifugal pumps of 200 gallons capacity per minute each, driven by motors of 7.75-horsepower each. Branches are led as required from the main to the various lavatories, pantries, etc., and the small gravity tanks enumerated above.

REFRIGERATING PLANT.

The ice-machine room is on the second platform, port side, between the engine and fire rooms. The plant consists of the following apparatus:

2 Kroeschell, vertical, motor-driven, double-acting CO_2 compressors, each capable of producing the ice-melting effect of 6 tons of ice per day. Driving motors, 15-horsepower each.
2 Brine-circulating pumps of the horizontal, motor-driven, centrifugal type. Driving motors, 3-horsepower each.
2 Water-circulating pumps of same type and horsepower as the brine-circulating pumps.
1 Double-pipe, horizontal, CO_2 condenser.
1 Brine-cooling and storage tank.
2 9-can ice boxes.
1 Ice cream freezer.
1 400-quart hardening box for ice cream.
2 Scuttle-butt coils.
4 Cold-storage rooms.

The cold-storage compartment is located on the third deck, immediately above the ice-machine room, and consists of four rooms, one each for meat, butter, officers, and the crew, the latter serving also as vestibule. The rooms are heavily insulated and fitted with cooling coils of 1¼-inch galvanized iron pipe. The following data is given for the cold-storage compartments:

Compartment.	Volume, cu. ft.	Cooling surface of coils, sq. ft.	Temperature to be maintained.
Meat room	2,100	960	15°F.
Butter room	152	90	40°F.
Crew's room	422	200	40°F.
Officers' room	460	230	40°F.

Except the ice boxes, which are direct cooled by CO_2 coils submerged in a brine bath surrounding the ice cans, the plant is operated on the brine-circulating system. The brine is cooled in the cooling tank, also provided with CO_2 coils, submerged in the brine, and then circulated through the system by the brine-circulating pumps. The return brine is led back to the cooling tank, where it is again cooled and the cycle repeated.

RESERVE FEED TANKS AND CONNECTIONS.

Double-bottom compartments B-1 to B-4, inclusive, are fitted up as reserve feed tanks.

The tanks are filled through a 4-inch pipe fitted with two 2½-inch hose connections at the ship's sides, port and starboard, and connected to the combined filling and suction manifold in fireroom No. 2.

There is also a 2½-inch filling pipe from the distiller freshwater main, which is led to the same manifold. From the manifold 4-inch combined filling and suction pipes are led to the bottom of each tank.

There is a 4-inch main between the manifold and the auxiliary-feed pump suctions, so that the auxiliary-feed pumps may draw from any tank. The auxiliary-feed pump in fireroom No. 2 is arranged to draw direct from any tank and discharge to any other tank. The reserve feed pump in the engine-room also has a 4-inch suction from the manifold, for drawing water direct from any tank and discharging same to the main feed tanks.

VENTILATING SYSTEM.

Artificial ventilation is provided where necessary for all quarters, living spaces, passages, storerooms, magazines, engine rooms, dynamo rooms, evaporator room, etc.

In general air is supplied on the plenum system to the different compartments requiring ventilation. The toilet spaces are also provided with the exhaust system.

The ventilating fans are of the B. F. Sturtevant Company's type, driven by G. E. Co. motors, except for a few minor exceptions.

Heater boxes are fitted in the ventilating ducts to all quarters below the main deck, crew's space, etc., for heating the incoming air in cold weather; no other heating apparatus being provided for these spaces.

HEATING SYSTEM.

All staterooms and quarters (except the captain's, admiral's and chief-of-staff's quarters), crew's space, etc., are heated by the thermo-tank ventilating system; heater boxes provided with suitable steam coils are placed in the air duct for heating the air supplied to these compartments. The heater boxes are of the Zimmerman type, as manufactured by the Schutte-Koerting Co. The remaining portions of the vessel, except the turrets, torpedo room and charthouse, which are provided with electric heaters, are heated by the customary pipe-coil steam radiators.

The heating system is divided into two main sections, one forward and one aft. The former takes steam from the auxiliary steam pipe in fireroom No. 1, and the latter from the auxiliary steam pipe in the center engine room. Stop and reducing valves are fitted at the branches from the auxiliary steam pipes.

The galley steam is taken off the auxiliary steam line in firerooms Nos. 2 and 3; a stop valve and a reducing valve are provided at each connection.

The drains from the heating system and galley are led to suitable traps, which discharge to the feed and filter tanks and condensers.

MAIN PROPELLING MACHINERY AND MAIN AND EXCITER SWITCHBOARDS.

The main propelling machinery of the *New Mexico* consists of two alternating-current turbo-generators, four induction motors, two 300 kw. direct-current generators for excitation and motor-driven auxiliaries, two motor generator boosters, a main switchboard, an exciter switchboard, ventilating blowers for the main motors, and the necessary wire and cable for connecting up the generators and motors to the switchboards.

The weight of the propelling machinery, exclusive of spare parts, is about 500 tons. This is about 215 tons less than the weight of the *Mississippi*, a sister ship of the *New Mexico*.

A section of the main turbine is shown in Figure 1. It is a 10-stage, G. E.-Curtis turbine. The first stage is velocity compounded, having two rows of moving buckets, and the other stages are pressure compounded, having a single row of moving buckets. The wheels are pressed on to the shaft and keyed. The tenth-stage wheel secures against a shoulder on the shaft and the first-stage wheel is secured by a cross key in the wheel and shaft, thus preventing fore and aft motion of the wheels. The wheels are separated approximately .01 inch by crushing pieces inserted between the wheels. The blades are secured to the wheels by dovetails, as shown in Figure 1. All stages except the first have complete peripheral admission. The admission nozzles are cast into diaphragms which separate the successive stages. These diaphragms are split through the horizontal plane and are secured in the casing against shoulders. The lower half of each diaphragm may be rolled out without lifting the rotors. When the casing is lifted, the top half of the diaphragm may be lowered by backing off a screw on each side which takes the

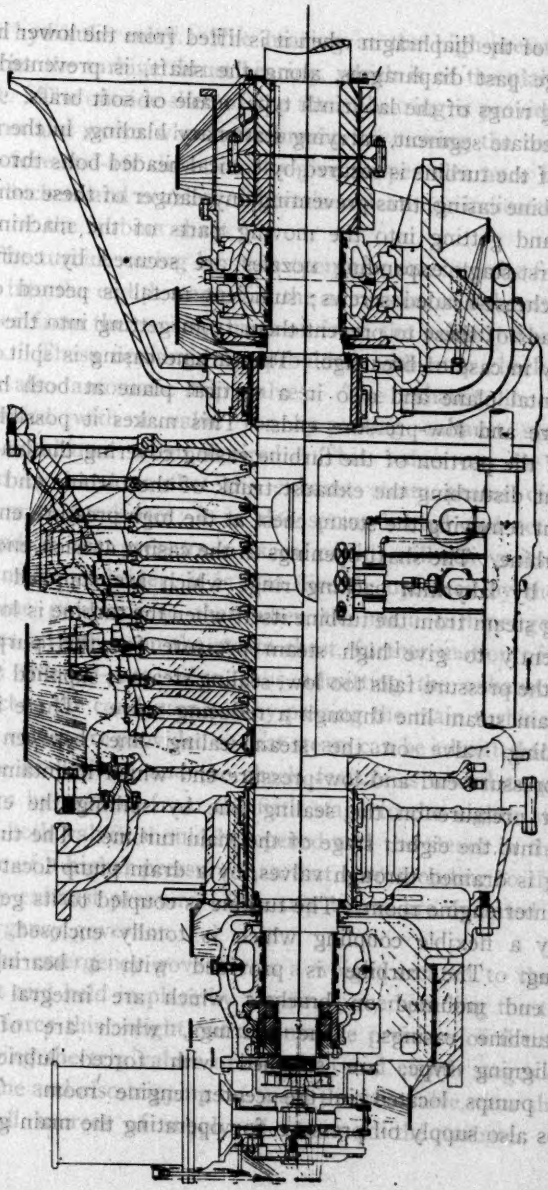


FIG. 1.—CROSS SECTION OF MAIN TURBINE.

weight of the diaphragm when it is lifted from the lower half. Leakage past diaphragms, along the shaft, is prevented by packing rings of the labyrinth type, made of soft brass. The intermediate segment, carrying stationary blading, in the first stage of the turbine is secured by conical-headed bolts through the turbine casing, thus preventing any danger of these coming loose and getting into the moving parts of the machinery. The first-stage expanding nozzles are secured by countersunk, cheese-headed screws; sufficient metal is peened over the heads of these to prevent them from getting into the machinery in case of breakage. The turbine casing is split in a horizontal plane and also in a vertical plane at both high-pressure and low-pressure ends. This makes it possible to lift off the portion of the turbine casing covering the blading without disturbing the exhaust trunk of the turbine and also without removing the steam chest at the high-pressure end of the turbine. The shaft openings in the casing at each end are sealed by labyrinth packing rings which are supplied with sealing steam from the turbine itself when the turbine is loaded sufficiently to give high steam pressure for this purpose; when the pressure falls too low, sealing steam is supplied from the main steam line through a reducing valve. There is an unloading valve on the steam-sealing line between the high-pressure end and low-pressure end which maintains the proper pressure on the sealing line by spilling the excess steam into the eighth stage of the main turbine. The turbine casing is drained through valves, by a drain pump located in the center engine room. The turbine is coupled to its generator by a flexible coupling which is totally enclosed by a housing. The turbine is provided with a bearing at each end mounted on brackets which are integral with the turbine casings. The bearings, which are of the self-aligning type, are supplied with forced lubrication from pumps located in the center engine room. These pumps also supply oil pressure for operating the main gover-

nor hydraulic relay. The bearing at the high-pressure end carries the turbine thrust bearing, which is of the plain collar type. The outside shell of the thrust bearing is provided with worm and worm wheel attachment for moving the turbine in the fore-and-aft direction, to give proper clearance between the buckets and nozzles. There is a clearance indicator at the end of the turbine shaft.

The turbine casing has two openings, one into the fifth and one into the eighth stage, for the admission of exhaust steam, either from the exciting units or from the exhaust line of the ship. These openings are provided with stop check valves and also automatic trip valves. The exhaust line is also provided with a valve for by-passing exhaust steam to the condenser when desired. This valve can be locked open by hand so that the exhaust steam passes direct to the main condenser; it is also spring-loaded and set at ten pounds so that pressure in excess of this will relieve itself into the main condenser; it is also directly operated by the main turbine governor so that when the main turbine governor closes all valves on the steam chest, further motion of it will open this by-pass valve, thus admitting the exhaust steam direct to the condenser and by-passing the main turbine. The stop valves are provided so that steam can be admitted to either stage as desired, and the check feature is added to prevent steam by-passing from the fifth stage to the eighth stage in case both valves should be opened and the steam in the fifth stage of a higher pressure than that corresponding to the auxiliary exhaust pressure. The trip valves are operated by the emergency governor.

The emergency governor is a weight secured to the main shaft and held in place by a spring; overspeeding the turbine will force this weight out against the pressure of the spring, tripping the trip valves of the fifth and eighth stages of the turbine and also tripping the main throttle valve, thus shutting off all sources of supply of steam to the turbine. These

valves can also be tripped by a hand pull located in the center engine room.

The main governor is driven by a worm gear from the high-pressure end of the turbine shaft. It is of the flyball type, the weights being mounted on knife edges and working against a spring. This governor is adjustable and can be set to give any desired speed from 700 revolutions to 2,200 revolutions, and when once set will maintain that speed regardless of changes in load.

A diagrammatic sketch of the governor and its control gear is shown in Figure 2. The governor weights operate a pilot valve of a hydraulic relay, which, in turn, admits oil pressure to a piston driving a rack; this, in turn, drives a pinion which is mounted on a camshaft; motion of these cams opens successive valves in the steam chest of the turbine, thus admitting the desired amount of steam. This control mechanism for the governor is moved from the center engine room by means of rods and bell cranks, as indicated in Figure 2. The effect of the motion of this transmission system is to shift the fulcrum about which the governor weights act and thus to give a different speed for each setting of the fulcrum. In addition to the system of bell cranks for setting the fulcrum, there is another system of bell cranks and rods coming from the center engine room which operates a stop shown diagrammatically in Figure 2. The effect of this stop is to limit the motion of the governor for any given setting, and thus limit the amount of steam that the turbine can take for that setting. This steam limit is necessary in connection with the maneuvering of the ship, as will be explained later on in this article.

A cross section of the main generator is shown in Figure 3, and the assembled rotor is shown in Figure 4. The generator is normally rated at 11,500 kw. at 80 per cent. power factor, and has an overload rating of 25 per cent. It is quarter-phase, two-pole, and designed to run at about 2,100 r.p.m.

The stator windings of the two phases are arranged as

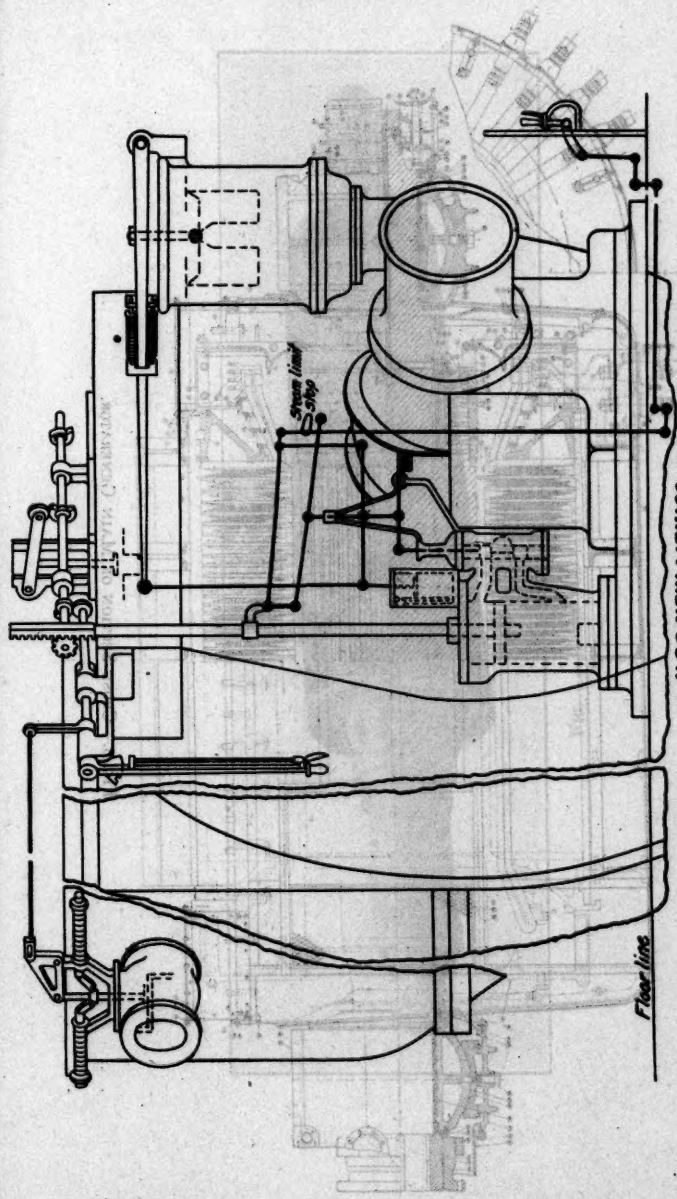

Steam Line Stop
Floor line
U.S.S. NEW MEXICO.
ARRANGEMENT OF CONNECTIONS OF OPERATING MECHANISM

Fig. 2.

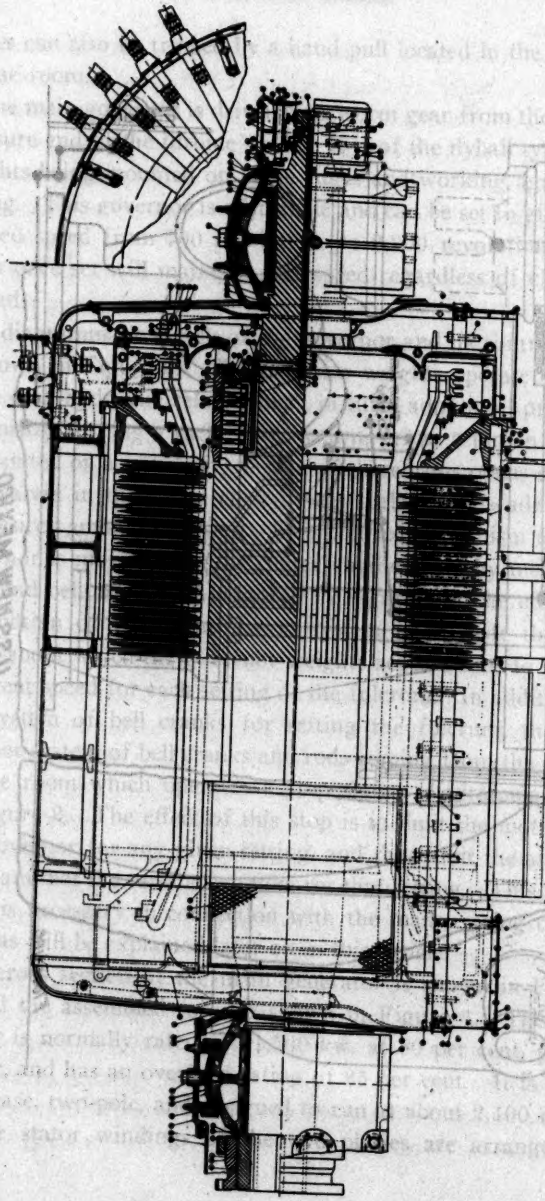


FIG. 3.—CROSS SECTION OF MAIN GENERATOR.

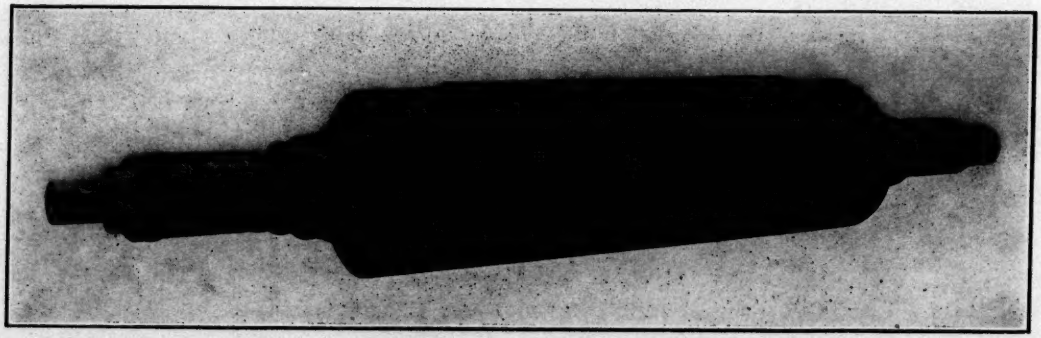


FIG. 4.—ROTOR FOR MAIN GENERATOR.

shown in Figure 5. Both ends of each independent winding are brought to the terminals of a double-throw switch; this switch, when in one throw, puts the winding in the diametral or parallel connection, and with the generator operating at designed speed and field strength, gives 3,000 volts; with the switch in the other throw, the windings are

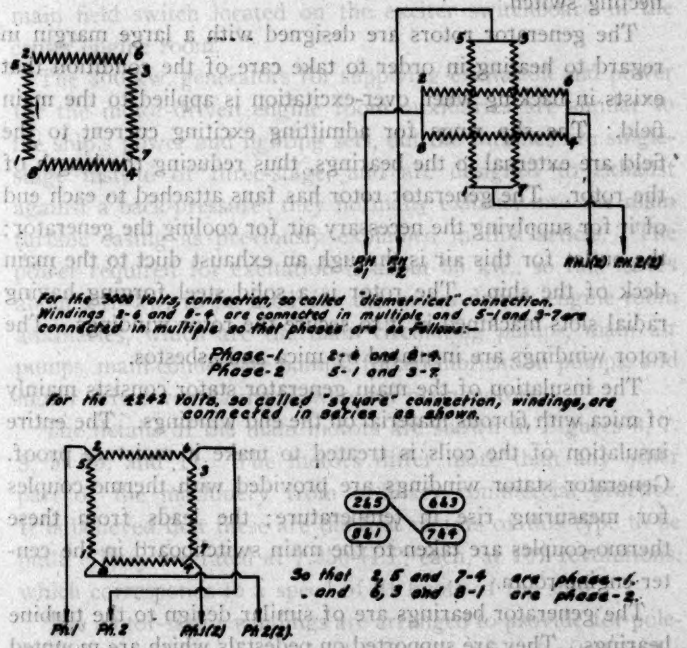

For the 3000 Volts, connection, so called "diametrical" connection.
Windings 2-6 and 8-4 are connected in multiple and 5-1 and 3-7 are connected in multiple so that phases are as follows:-
Phase-1 2-6 and 8-4
Phase-2 5-1 and 3-7
For the 4242 Volts, so called "square" connection, windings, are connected in series as shown.
245
663
841
784
So that 2, 5 and 7-4 are phase-1.
and 6, 3 and 8-1 are phase-2.
Ph.1 Ph.2 Ph.1(2) Ph.2(2)

FIG. 5.—MAIN GENERATOR CONNECTIONS.

thrown into the square or series connection and this arrangement at designed field and speed of the generator gives 4,242 volts. The windings are arranged in this way so as to give the maximum efficiency of the generator when one generator is driving four motors and also when it is driving only two motors. In the first case the low-voltage connection is used and this utilizes the increased current path when using four

motors. When using the high-voltage connection, only two motors are driven from one generator, thus cutting the current path down to one-half its value in the other case. The switch for making this change in "set-up" is mounted on the outboard bulkhead of the center engine room and is operated in the center engine room, and is called the generator disconnecting switch.

The generator rotors are designed with a large margin in regard to heating in order to take care of the condition that exists in backing when over-excitation is applied to the main field. The slip rings for admitting exciting current to the field are external to the bearings, thus reducing the length of the rotor. The generator rotor has fans attached to each end of it for supplying the necessary air for cooling the generator; the outlet for this air is through an exhaust duct to the main deck of the ship. The rotor is a solid steel forging having radial slots machined in it to secure the rotor windings. The rotor windings are insulated by mica and asbestos.

The insulation of the main generator stator consists mainly of mica with fibrous material on the end windings. The entire insulation of the coils is treated to make it moisture proof. Generator stator windings are provided with thermo-couples for measuring rise in temperature; the leads from these thermo-couples are taken to the main switchboard in the center engine room.

The generator bearings are of similar design to the turbine bearings. They are supported on pedestals which are mounted on foundations built in the ship and are not attached to the generator casing.

Excitation for the generator is provided by the 300-kw. generator sets located in the center engine room and by motor-generator boosters. The field current comes from the 300-kw. generator and passes in series through the armature of the generator of the booster. Therefore, if the voltage of the booster generator is varied, the voltage imposed on the slip

rings of the generator field will also be varied. The booster generator is arranged so that its voltage can be either added to or subtracted from that of the 300-kw. generator, thus giving the desired range of field control of the main generator without changing the voltage of the 300-kw. generators. The field of the main generator is opened and closed by the main field switch located on the exciter switchboard in the center engine room.

The 300-kw. generators for supplying excitation and power for the motor-driven engine room auxiliaries are similar to the ship's power and lighting sets, but the turbines are single-stage instead of three-stage, and are designed to exhaust against a back-pressure; they normally exhaust into the main turbine casing as previously explained in this article. The power required for excitation is about 55 kw., so this leaves about 245 kw. to be used for the motor-driven engine-room auxiliaries, which are the main circulating pumps, main air pumps, main condensate pumps, forced-lubrication pumps, and motor-driven ventilating blowers for main motors.

The details of the main motors are shown in Figures 6, 7, 8, 9, 10, and 11. The motors differ more than any other part of the machinery from ordinary commercial practice. It is believed that these are the first motors of this type to be built. They are rated at 7,250-H.P. each, at 167 revolutions, which corresponds to a speed of 21 knots of the ship.

The motor-stator windings are arranged to provide for pole-changing, and there are pole-changing switches located in the center engine room. One throw of these switches will give 24 poles on the stator of the motor and the other throw of the switch will give 36 poles on the stator of the motor. This gives in the first case a 12 to 1 reduction of turbine speed on the propeller, and in the second case gives an 18 to 1 reduction between turbine speed and propeller speed. This enables the turbine to be operated at full speed, both at 21 knots and at 15 knots, and adds largely to the economy of the turbine.

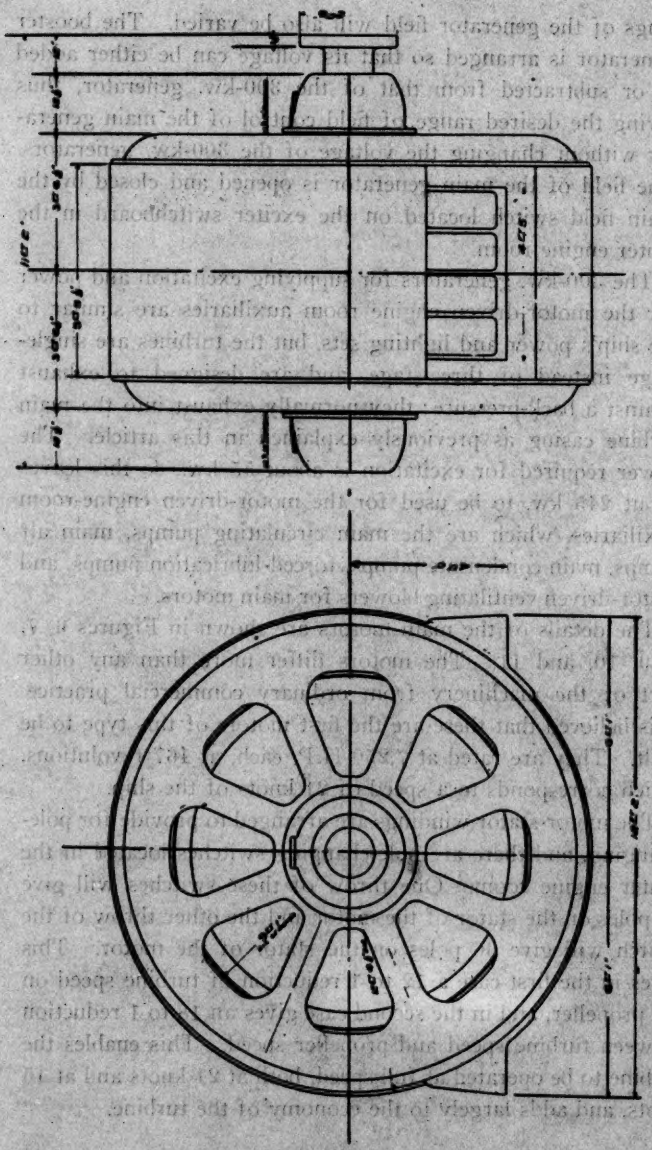


FIG. 6.—OUTLINE OF MAIN MOTOR.

FIG. 6.—OUTLINE OF MAIN MOTOR.

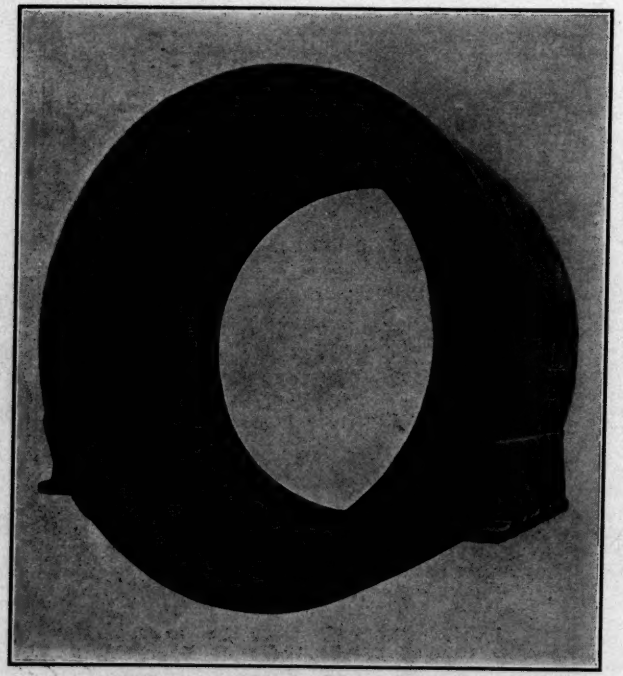


FIG. 7.—STATOR FOR MAIN MOTOR.

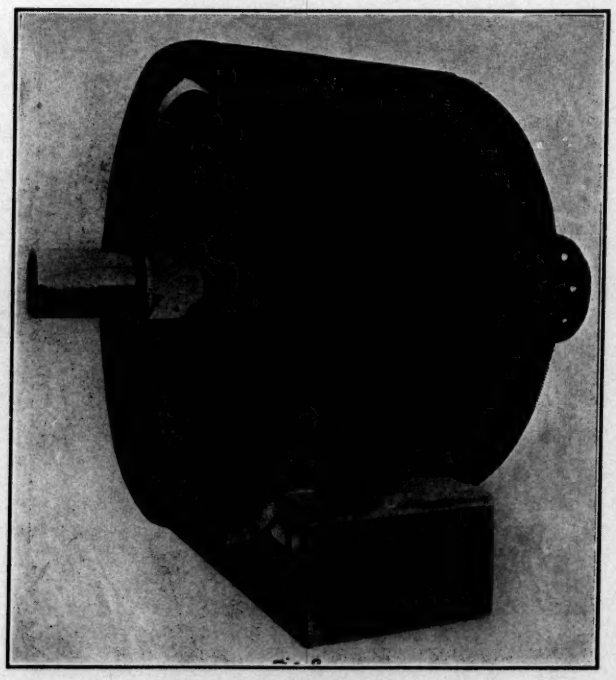


FIG. 8.—ROTOR FOR MAIN MOTOR.

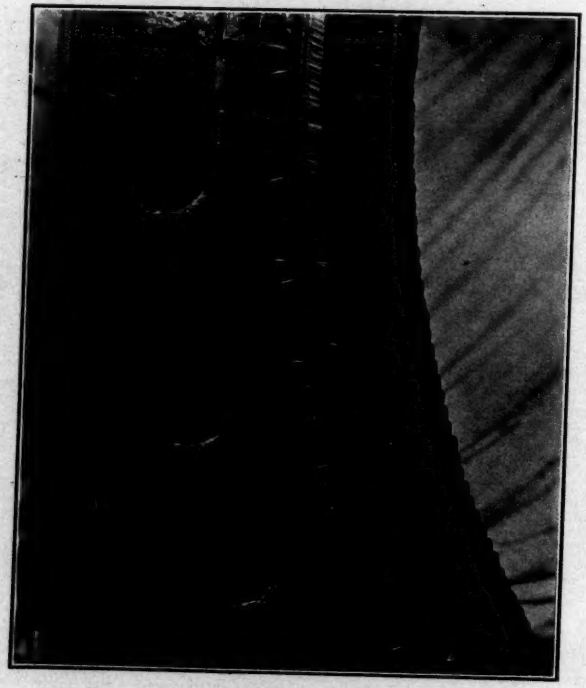


FIG. 9.—STATOR WINDING OF MAIN MOTOR.

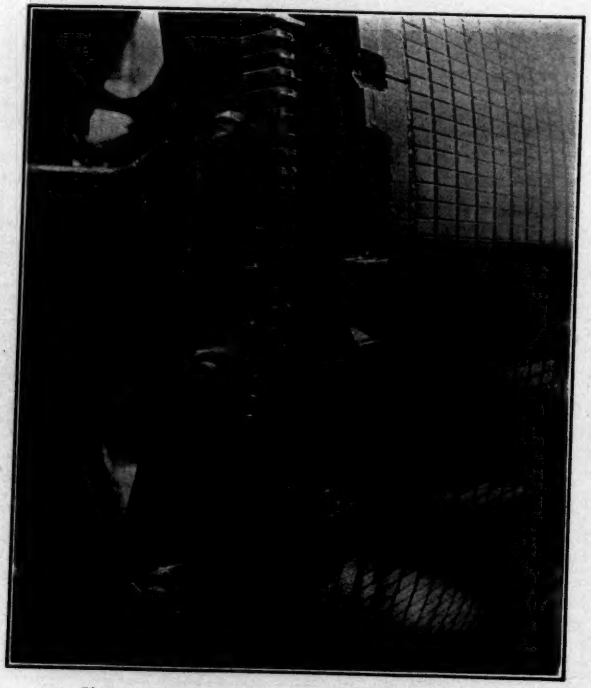


FIG. 10.—ROTOR WINDING OF MAIN MOTOR.

The stator and stator windings are shown in Figures 7 and 9. The insulation is similar to that of the generator stator, consisting mainly of mica, except for the end windings, which are provided with fibrous insulation, the whole being treated to make it moisture proof.

The motor rotor is shown in Figures 8 and 10. The rotor winding is a double squirrel-cage winding. Each slot in the rotor consists of one slot near the periphery of the rotor and one deeper in the iron of the rotor, the two being connected by a long, narrow air gap, as shown in Figure 11. Each of

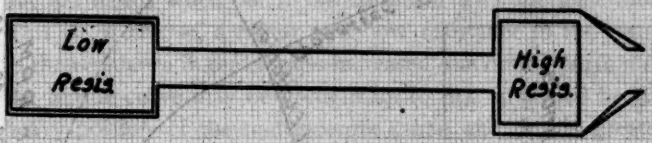


FIG. 11.—ROTOR SLOT.

these slots is provided with an indpendent squirrel cage. The squirrel cage in the outer slot is composed of high-resistance bars and that of the inner slot is composed of low-resistance bars. The high-resistance bars are made of German silver and the low-resistance bars of copper. The bars are wedged tightly into the slots in both cases without insulation. Each squirrel cage has the end of its bars connected by short-circuiting rings; in the case of the high-resistance bars this short-circuiting ring is divided into segments connected by flexible copper connections to provide for expansion, as shown in Figure 10. The object of this construction of the rotor is to provide a winding which will give the necessary torque required for reversing and at the same time make it possible to use the squirrel-cage motor, which is a very simple type of motor with no insulation in its rotor and requiring no changes in its rotor winding when shifting from 36 poles to 24 poles, and vice versa, in the motor stator.

FIG. 10.—ROTOR WINDING OF MAIN MOTOR.

FIG. 9.—STATOR WINDING OF MAIN MOTOR.

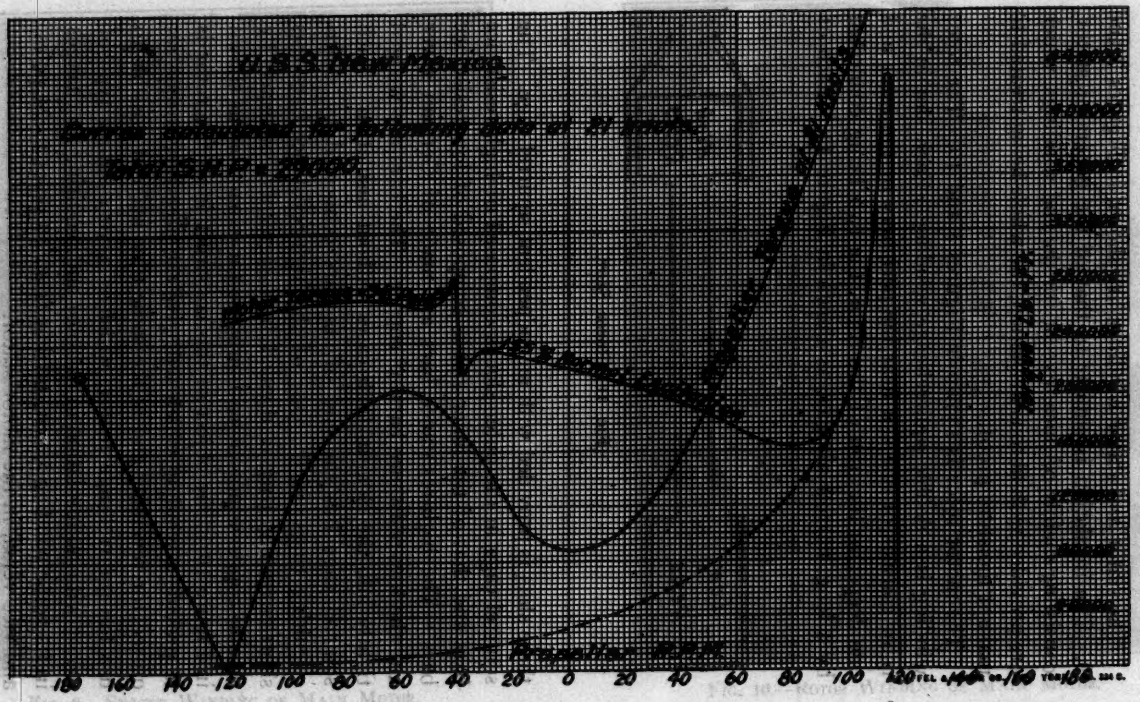


FIG. 12.—MOTOR AND PROPELLER TORQUE CURVES.

To better understand the requirements of the induction motor for backing, it will be necessary to refer to Figure 12, which gives the shape of both propeller-torque curve and motor-torque curve. It will be seen that the maximum torque required during the process of bringing a propeller to rest, with the ship going ahead, is almost as great as that required to drive the ship ahead at full speed. This is a very severe condition for induction motors to meet, as the torque of an induction motor when "out of step" or out of synchronism with its generator is very small indeed. The dotted portion of the motor-torque curve shows the torque of an ordinary induction motor when out of synchronism with the generator. This is, of course, a motor with low resistance in the rotor. If the resistance in the rotor is increased, the dotted portion of the torque curve may be brought up to that indicated by the heavy portion of the motor-torque curve and this is greater than the propeller-torque curve and will therefore reverse the propeller. It will be seen that the motor and propeller curves intersect each other in the given Figure at about 48 revolutions, or while the motor is still out of synchronism with the generator. It would be impossible to further increase the speed of the motor under this condition, as its torque begins to fall off with increase of speed, but by slowing down the turbine this point of intersection can be moved along on the propeller-torque curve until the propeller-torque curve intersects the motor-torque curve in that portion of the curve to the right of the maximum point, or in the "in synchronism" part of the curve. After the motor has come into synchronism with its generator the two can then be brought up together to whatever speed is desired. The increase in the resistance of the rotor which brings about the increase in the torque of the motor in the "out of synchronism" part of its curve may be best understood by reference to Figure 13. This diagram is an equivalent circuit to that of the double squirrel-cage induction motor. It will be noted that part of the reactance of the two

FIG. 12.—MOTOR AND PROPELLER TORQUE CURVES.

squirrel cages is mutual and part of it pertains only to the lower squirrel cage. At low frequencies both of these reactances are very small and have no great effect on the current flow, but at high frequencies they become very large and that of the inner squirrel cage becomes very much larger than the outer, thus causing the greater part of the current to flow

X_p = reactance of primary winding.
R_p = resistance of primary winding.
X_c = mutual reactance of the two squirrel cages.
g_m = magnetizing susceptance.
R_c = resistance of upper squirrel cage.
R_a = resistance of lower squirrel cage.
X_a = reactance of the lower squirrel cage, which is not mutual with the upper squirrel cage.

FIG. 13.—EQUIVALENT DIAGRAM OF DOUBLE SQUIRREL-CAGE MOTOR.

through the outer or high-resistance squirrel cage and giving the equivalent of a high-resistance squirrel-cage motor. As the frequency falls, this reactance decreases in value until at the low frequencies in the rotor, such as obtain when going ahead, the reactance virtually disappears and we have the equivalent of an efficient low-resistance squirrel-cage motor. This winding, therefore, gives automatically the equivalent of a high-resistance squirrel cage for backing and a low-resistance squirrel-cage for driving under normal conditions. The physical explanation of what takes place when the motor-stator windings are reversed, thus giving double generator frequency in the rotor, is that the flux in the rotor which normally encircles both squirrel cages is forced across the narrow air gap, practically short-circuiting the low-resistance bar and leaving it out of action almost entirely; as the frequency

falls, due to the reversal of the motor and its coming up to synchronism, more of this flux encircles the lower bar until the low frequencies are reached, when practically all of it again passes around the lower bar. The reversal of the

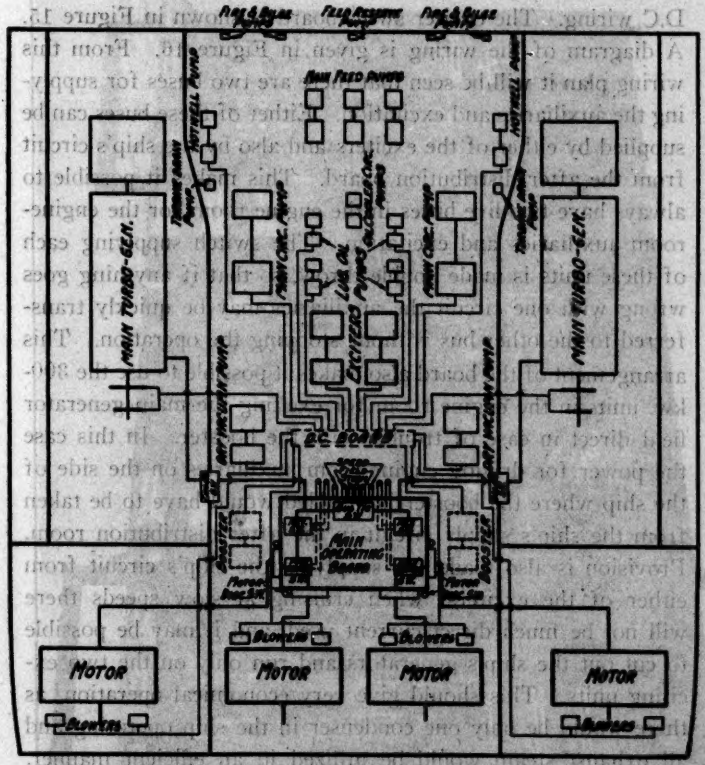


FIG. 14.—ARRANGEMENT OF MACHINERY.

motors is accomplished simply by crossing two of the leads of one phase of the motor, and this is accomplished by reversing switches located in the center engine room.

The main motors are ventilated by suction blowers located one on each side of each motor on top of its casing; these

draw air through the motor and discharge into ducts which are carried up to the atmosphere on the main deck.

The general arrangement of the machinery in the three engine rooms and the motor rooms is shown in Figure 14, which also gives a single line wiring diagram of the main A.C. and D.C wiring. The exciter switchboard is shown in Figure 15. A diagram of the wiring is given in Figure 16. From this wiring plan it will be seen that there are two buses for supplying the auxiliaries and excitation. Either of these buses can be supplied by either of the exciters and also by the ship's circuit from the after distribution board. This makes it possible to always have two live buses in the engine room for the engine-room auxiliaries and excitation. The switch supplying each of these units is made double throw, so that if anything goes wrong with one circuit the auxiliaries may be quickly transferred to the other bus without stopping the operation. This arrangement of the board also makes it possible to use the 300-kw. units in the engine room for exciting the main generator field direct in case of trouble with the booster. In this case the power for driving engine-room auxiliaries on the side of the ship where the booster is disabled would have to be taken from the ship's supply circuit in the after distribution room. Provision is also made for supplying the ship's circuit from either of the exciters; when cruising at slow speeds there will not be much direct current used, and it may be possible to cut out the ship's generators and run only on the two exciting units. This should give very economical operation, as there would be only one condenser in the ship operating and all exhaust steam would be utilized in an efficient manner, either in the main turbine or feed heaters, or evaporators.

The exciter switchboard, in addition to the other direct-current switches mounted on it, has also the main field switch, which is a solenoid-operated switch, and is controlled from the main switchboard. This switch is provided with magnetic blow-out to reduce the arcing on opening the main field,

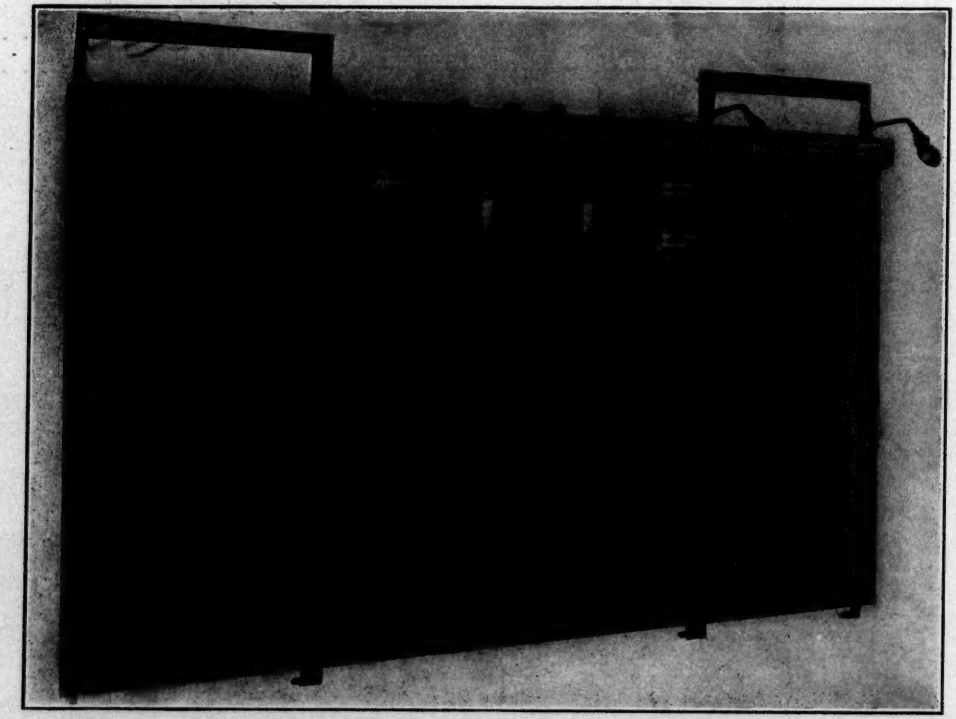
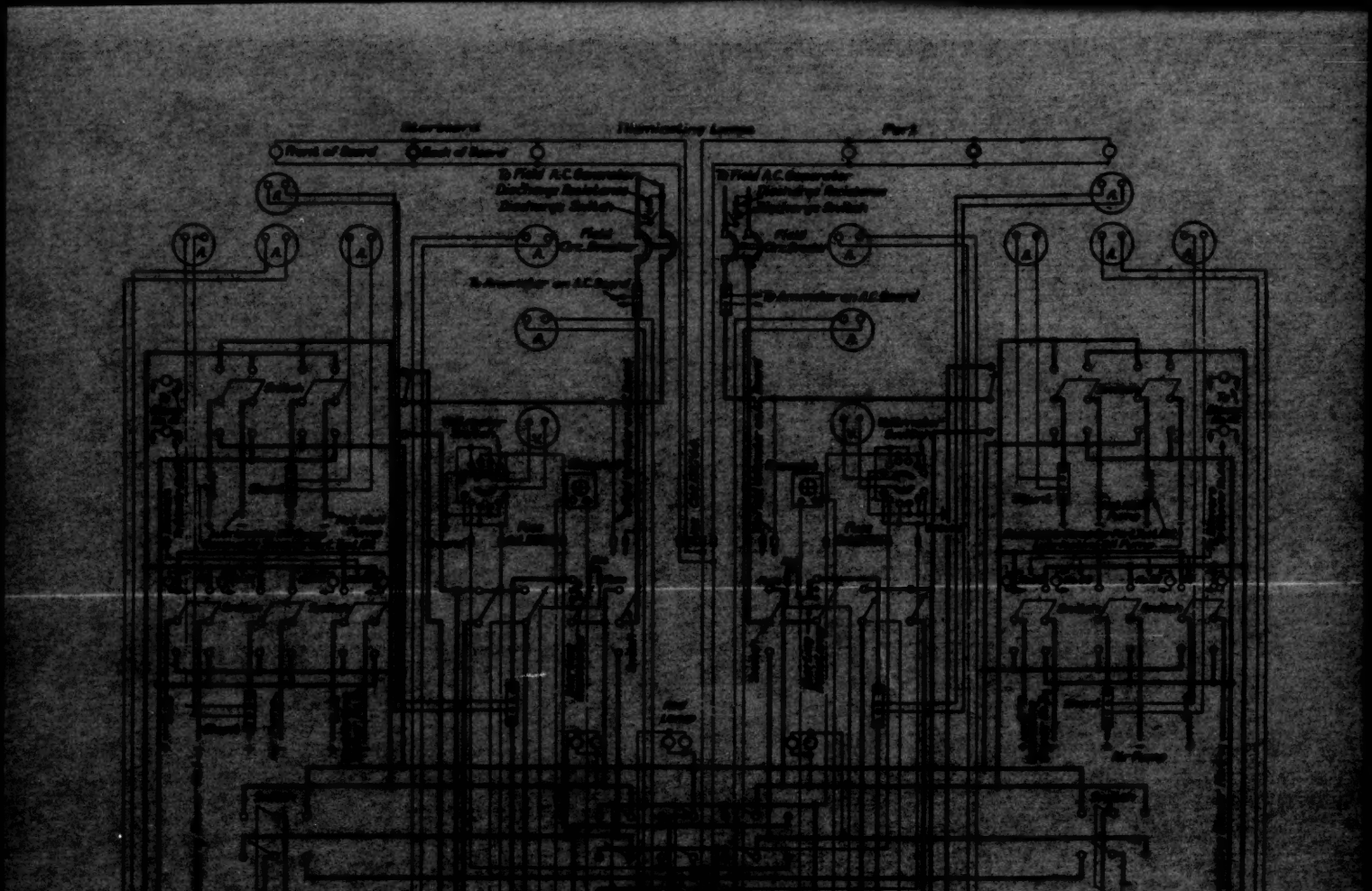
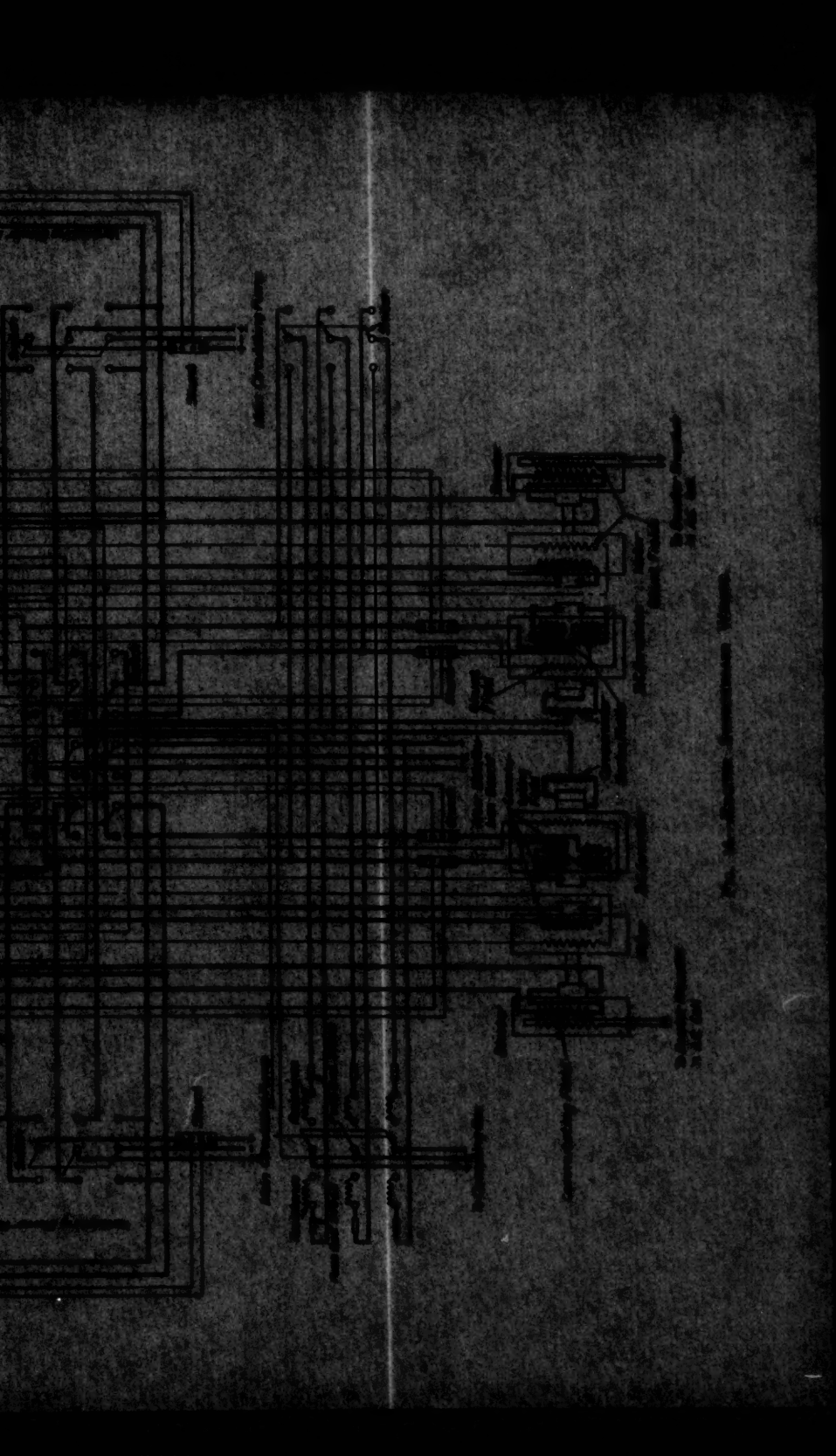


FIG. 15.—EXCITER SWITCHBOARD.





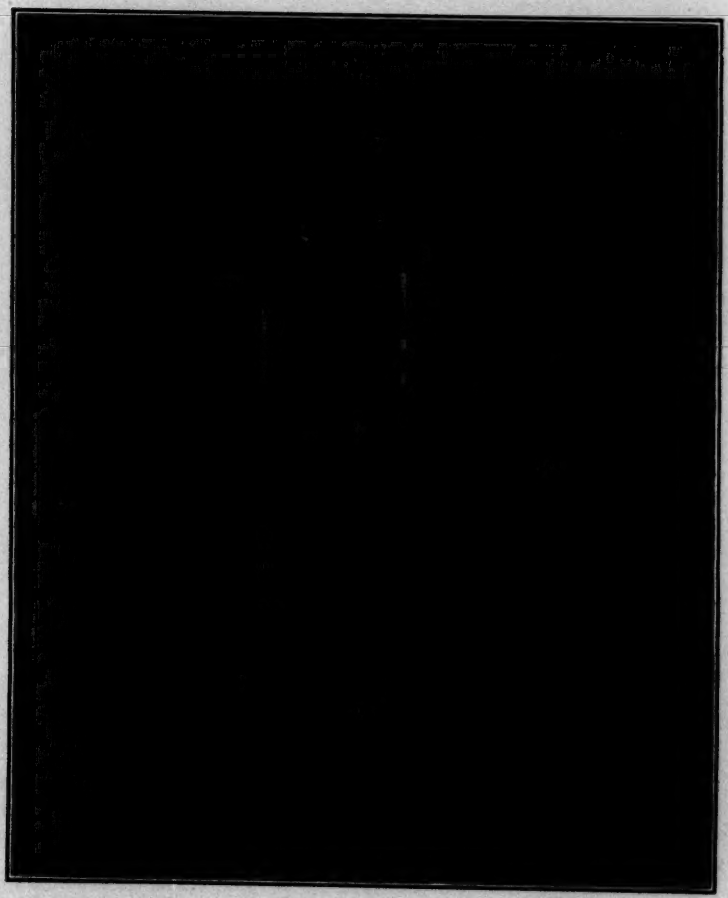


FIG. 17.—MAIN SWITCHBOARD—BACK VIEW WITH GRILLE WORK REMOVED.

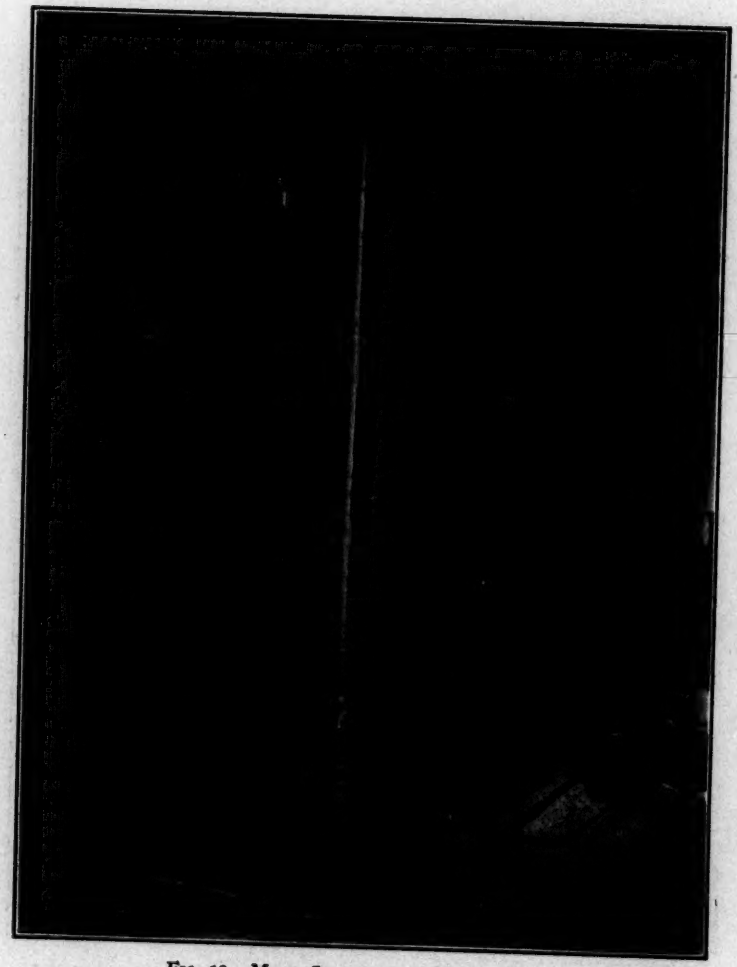


FIG. 18.—MAIN SWITCHBOARD—BACK VIEW.

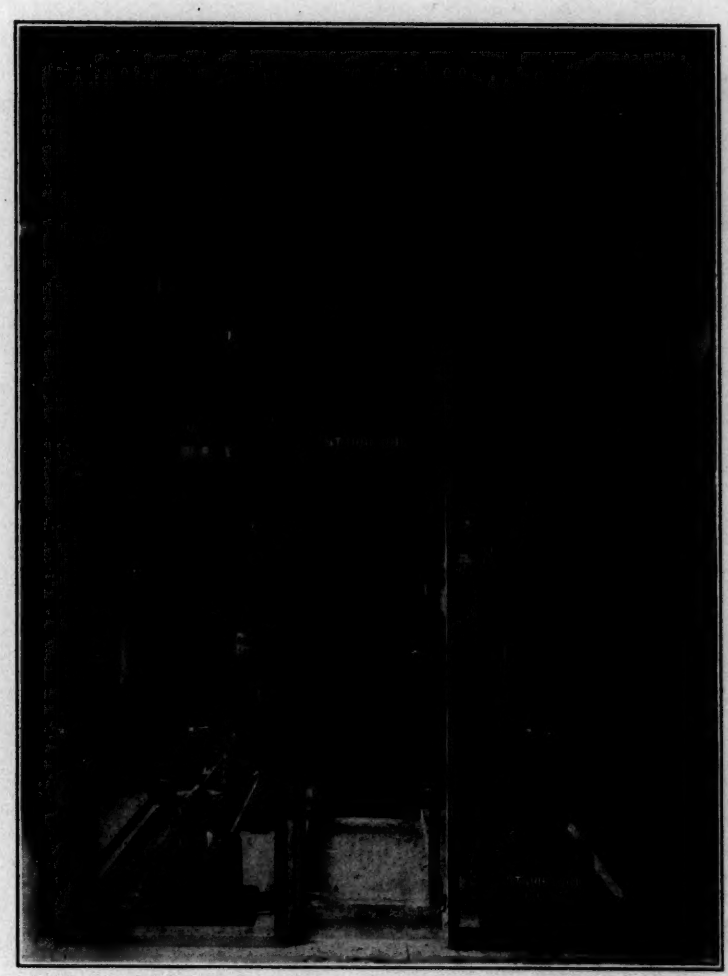


FIG. 19.—MAIN SWITCHBOARD—SIDE VIEW WITH GRILLE WORK REMOVED.

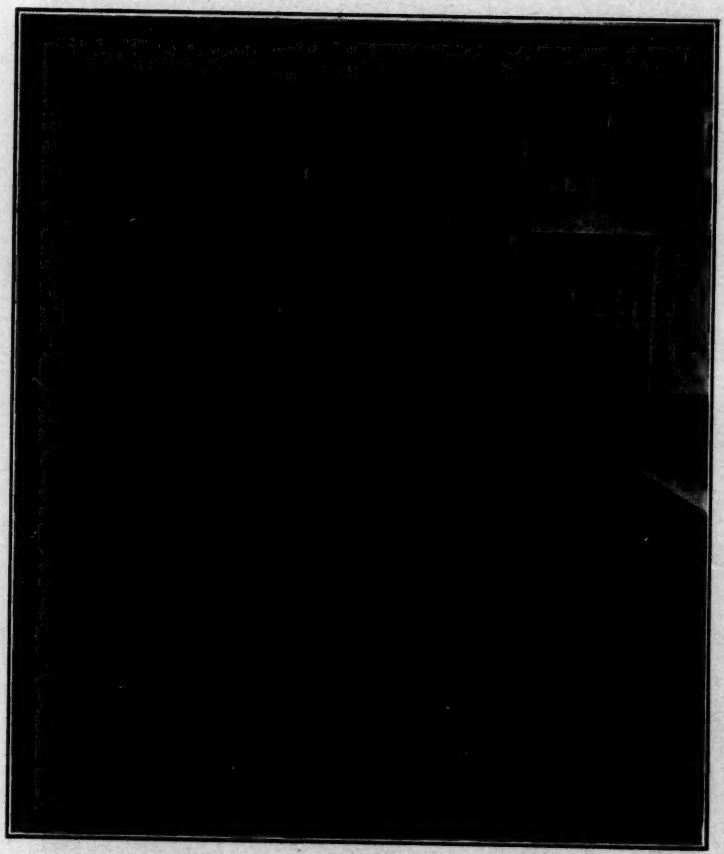


FIG. 20.—MAIN SWITCHBOARD—VIEW OF INTERIOR SHOWING INTERLOCKS AND OIL SWITCHES.

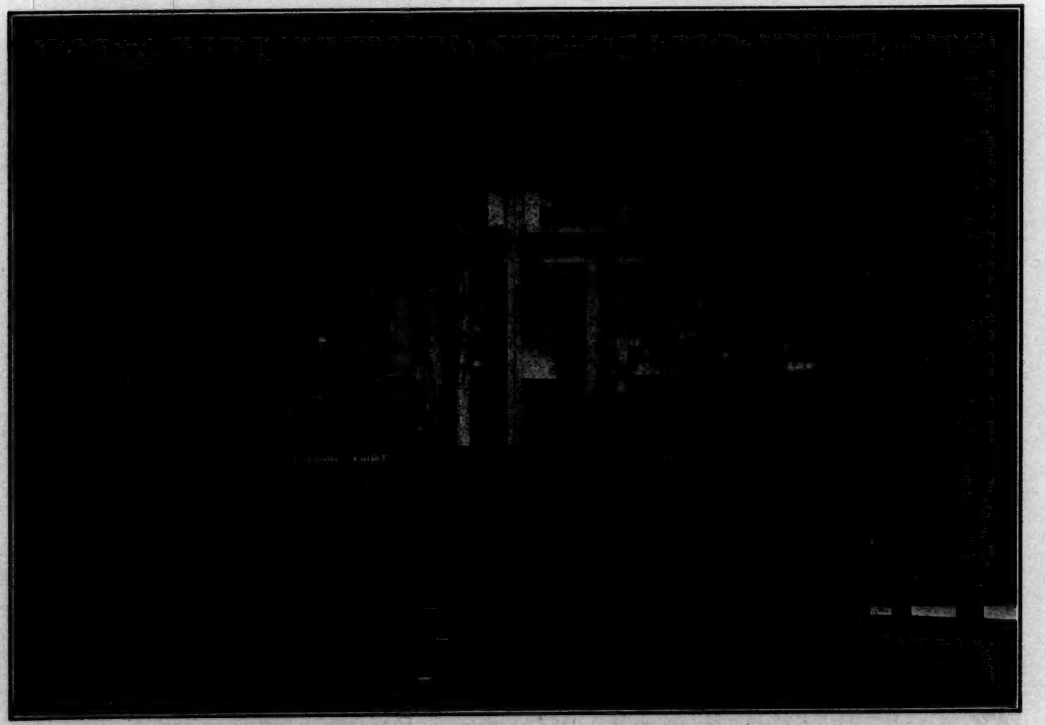


FIG. 21.—MAIN SWITCHBOARD—FRONT VIEW WITH PANELS REMOVED.

FIG. 22.—MAIN SWITCHBOARD—PARTIAL FRONT VIEW SHOWING PRINCIPAL CONTROL LEVERS.

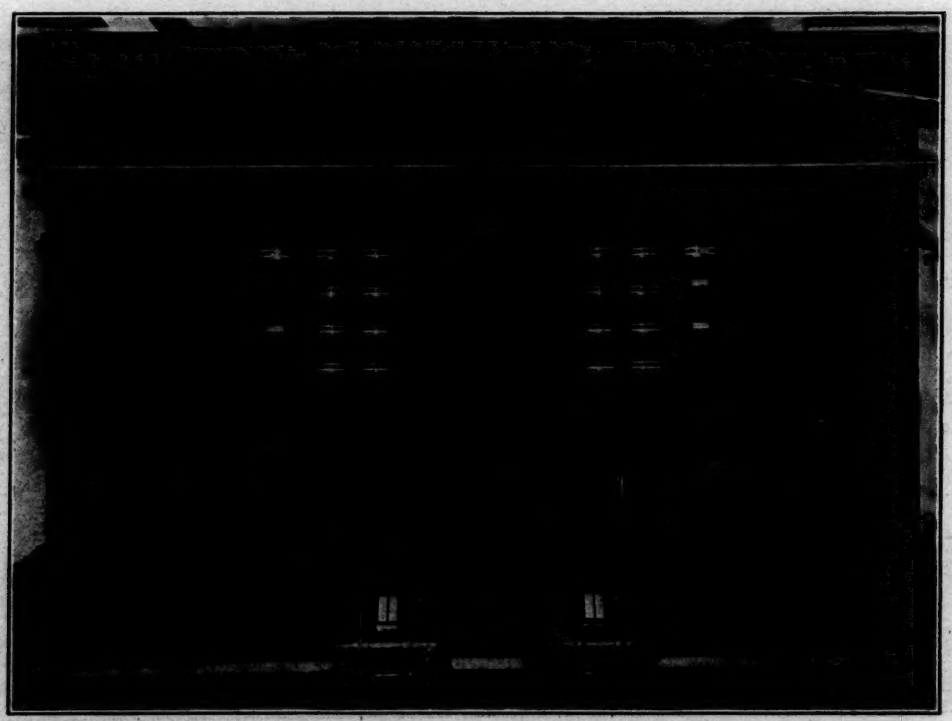


FIG. 23.—MAIN SWITCHBOARD—FRONT VIEW SHOWING LEVERS AND PRINCIPAL INSTRUMENTS.

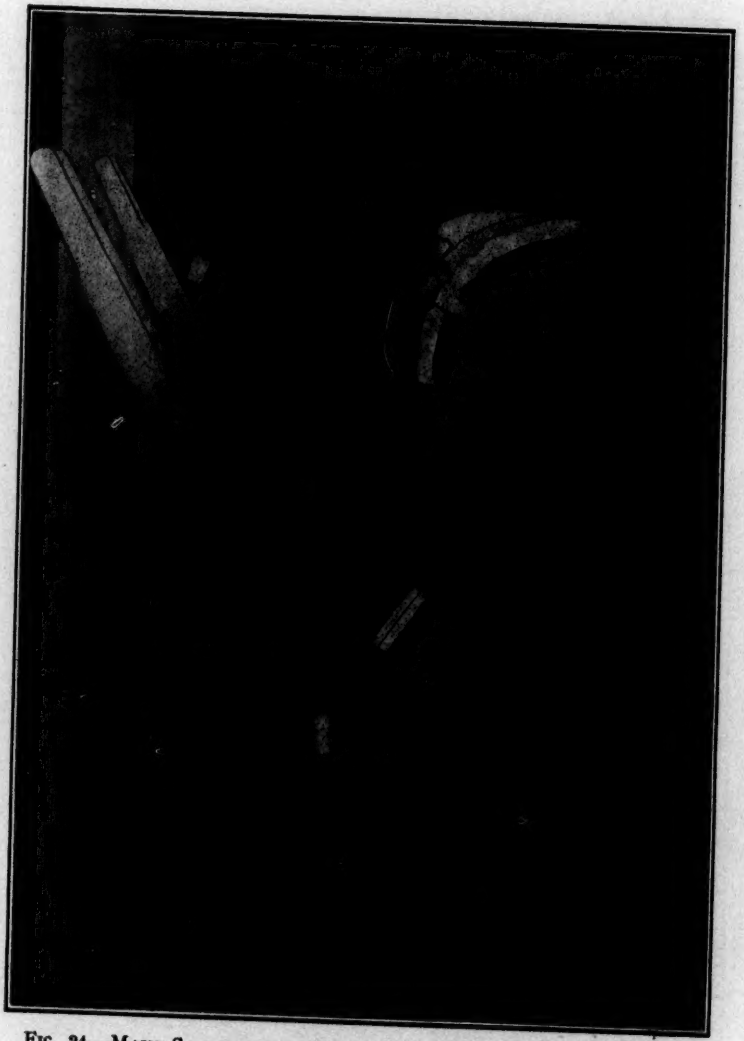


FIG. 24.—MAIN SWITCHBOARD—PARTIAL VIEW SHOWING CONTROL LEVERS.

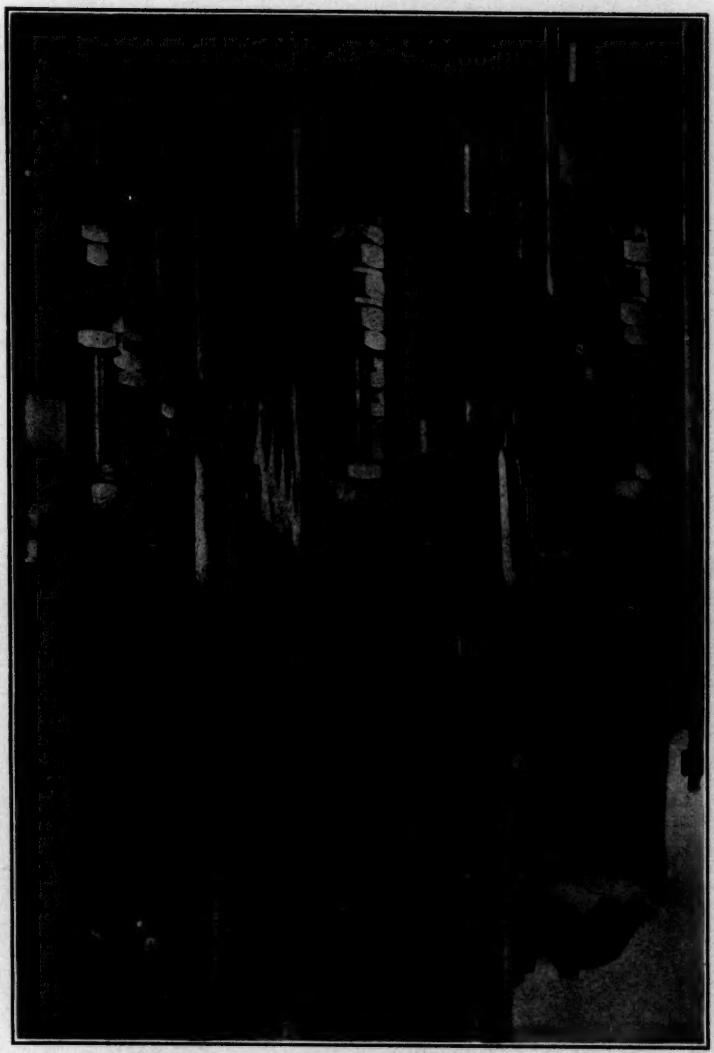


FIG. 25.—OIL SWITCH FOR REVERSING ON POLE CHANGING.

and is also provided with an auxiliary switch which short-circuits the main field through a discharge resistance, previous to opening the main field. This prevents any great rise of potential in the main field circuit due to the quick opening of the main field switch; if this resistance were not provided, opening of the main field switch would burn up the switch and also probably puncture the insulation of the main field itself. This switch is arranged for hand operation in case of failure of the solenoid-operated feature. It is also arranged so that it can be tripped out by a balanced relay located on the main switchboard and the purpose of which will be explained later on in this article.

Other switches on this switchboard are of the knife-blade type.

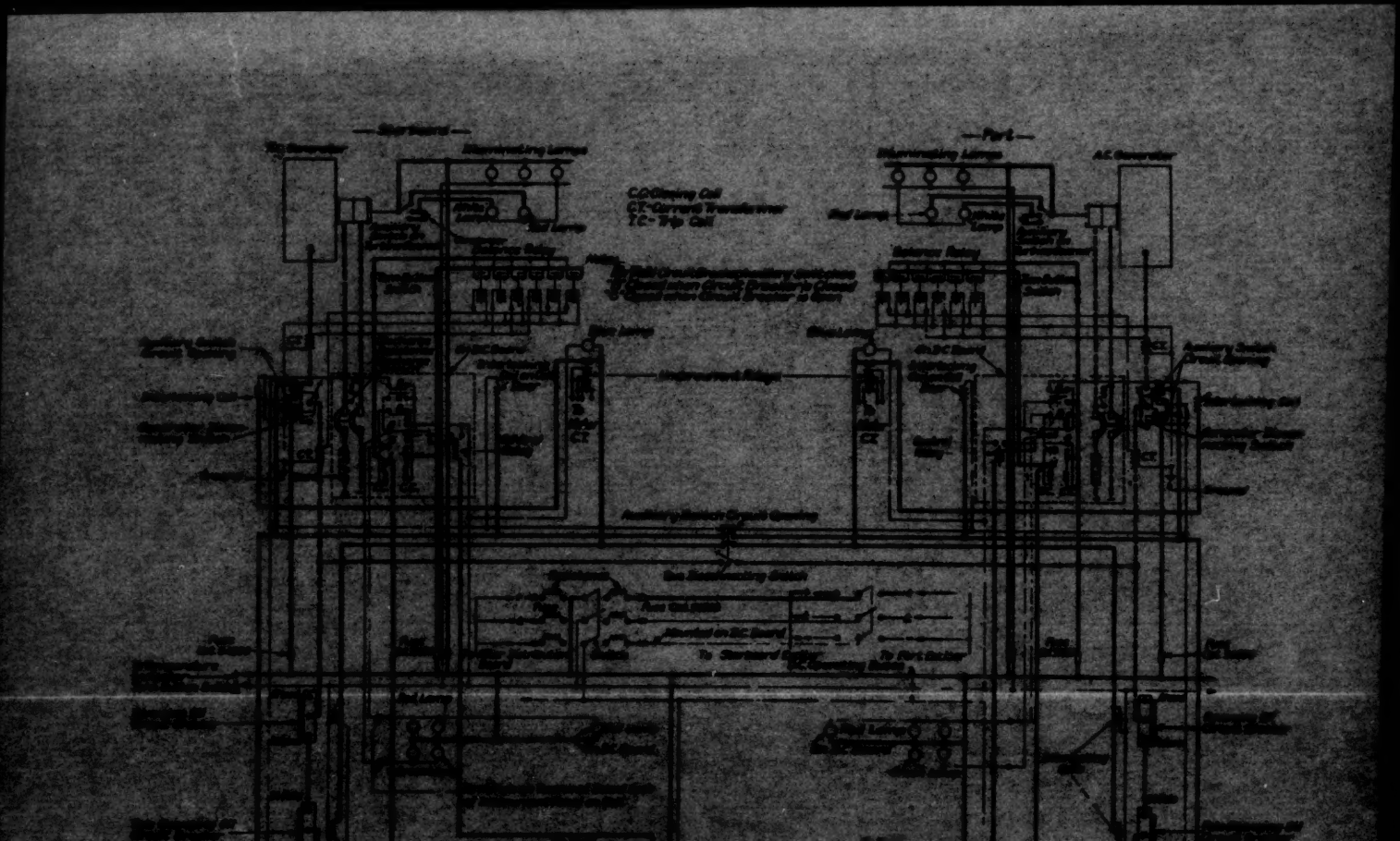
Indicating lamps are placed to show which bus is alive, thus facilitating the operation of the switching and lessening chances of mistake.

The main exciter supply switches are provided with interlocks, which prevent them from being thrown in parallel on the same bus or in parallel with the ship's circuit.

The main switchboard and the main A.C. wiring and D.C. interlock wiring are shown in Figures 17 to 28, inclusive. Figures 17 to 21, inclusive, show various views of the structure of the main switchboard and of the arrangement of disconnecting switches and the interior of the board itself, together with the mechanical interlocks between switches. From these views it will be seen that the main switchboard is a cell structure, built up on angles and carrying panels for mounting instruments on its forward face. The remaining part of the switchboard is enclosed with grille work and has a door at each end which gives access to the passageway down the middle of the cell. This passageway gives access to all mechanical interlocks located inside the structure, to all oil switches, and to the bus tie switch; the latter is the only switch that is actually operated from the inside of the cell

structure. Figures 22, 23 and 24 show various views of the operating levers on the front of the switchboard. Figure 28 shows the front of the main switchboard with all instruments and levers. All the levers used in the operation of the engines are shown in these views. The switchboard is located in the after end of the center engine room, and the operator faces aft when handling the switches. By referring to Figure 27 the relation of all the switches and levers used in the various operations can be seen. There is a disconnecting switch (already described under generators) for each main generator. These are operated in the center engine room and are mounted on the outboard bulkheads on each side. These switches are of the knife-blade type; they are mounted on large brass plates which are bolted to the bulkhead and form a part of it. Single conductor cable is used to connect the generator to the main switchboard, and it is therefore necessary to replace the part of the bulkhead surrounding the cable with non-magnetic material to prevent its overheating and causing losses in the circuits. The insulators for this switch are heavy molded blocks of bakelite, which is very tough and is not susceptible to breakage by shock or vibration.

The bus tie switch is mounted inside the cell structure and is operated at that point. It is also of the knife-blade type. The bus tie switch connects the starboard and port buses so that either generator can run all four motors when desired. The bus tie switch and the two generator disconnecting switches are mechanically interlocked in such a manner that only two of them can be thrown in at any one time. This makes it impossible to connect the two generators to the same bus and thus put them in parallel. They are also mechanically interlocked so that the generator disconnecting switches can be thrown into the low-voltage position only when the bus tie switch is closed, and into the high-voltage position only when the bus tie switch is open; this interlocking makes it certain that the generators will always be operated in the most effi-



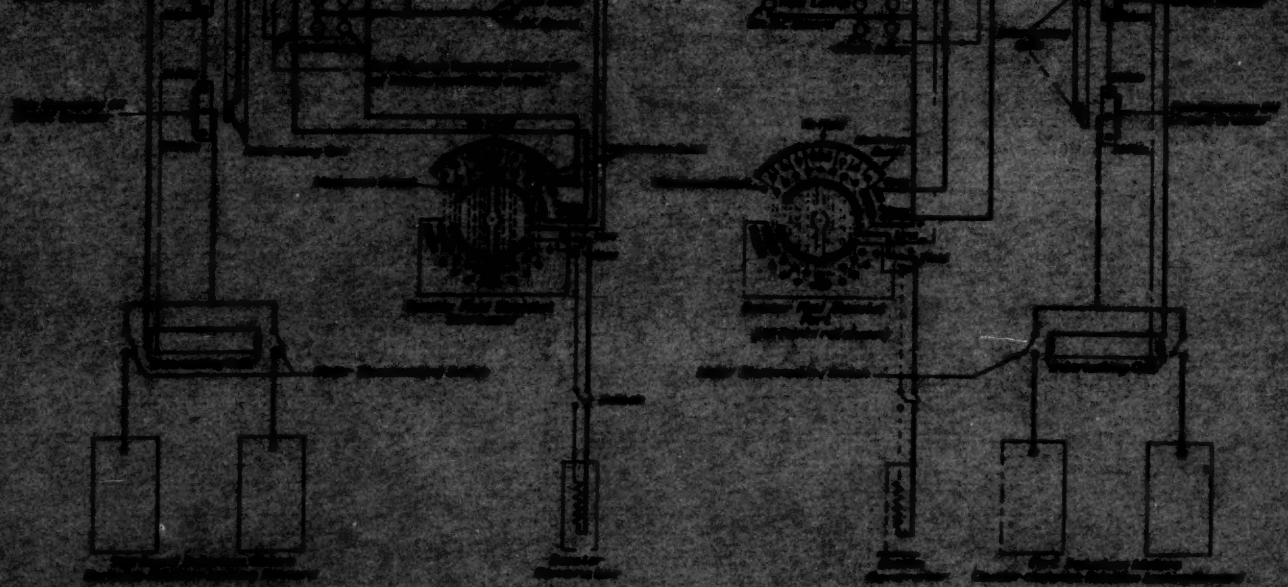
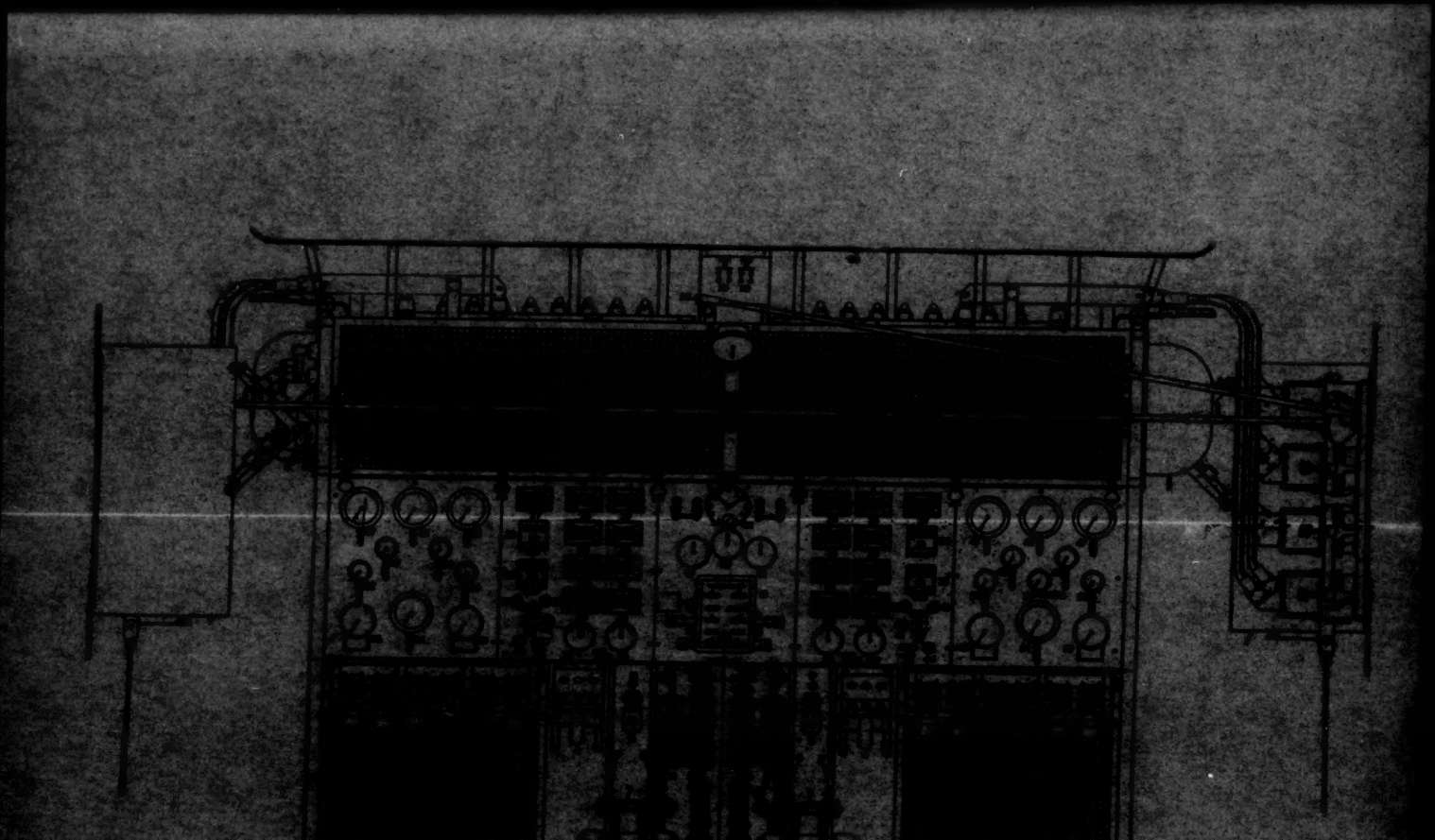


Fig. 36.—Carbon arc Interior Wiring.



cient way, as has already been described under the subject of generators.

The motor-disconnecting switches are mounted at the top of the cell structure on each side and at the back. These switches and the levers for operating them are shown clearly in Figure 18. These switches are also of the knife-blade type.

None of the knife-blade switches are to be operated when alive, and the means for preventing this will be explained later on in this article when the subject of electrical interlocks is discussed.

The reversing switches are of the laminated brush type, and are oil-break. They have auxiliary contacts and are intended to be large enough to break full-load currents, but in the actual operation this is never done, as will be explained under "Operation." The ahead and astern switches are mounted in the same oil tank. This switch is shown in Figure 25. It will be seen that there is a brush on each side, the two arms being pivoted at the center. The arrangement of the levers is such that one of these arms is moved up and the other down at the same time, one closing for the ahead direction and when moved in the opposite way the other closes for the astern direction. The function of this switch is merely to reverse the connection of two leads (one phase) to the motor. A third lead is, however, interrupted by this switch, as otherwise, opening this switch would still leave one phase connected to the motor, which would then run as a single-phase motor. The fourth lead to the motor passes direct from the bus to the motor and is not interrupted by the reversing switch. The reversing switches for the two motors on one side of the ship are operated by the same lever on the front of the switchboard.

The pole-changing switches are in every respect similar to the reversing switches. Contact of the laminated brushes on one side connects the motor for 24 poles and contact on the other side for 36 poles. The switches for the two motors on

one side of the ship are operated by one lever on the front of the switchboard.

The pole-changing and reversing switches are mechanically interlocked with the field lever so that they cannot be moved unless the field lever is in the open position. They are also interlocked with each other so that the reversing lever cannot be thrown in the astern position except when the pole-changer is in the 36-pole position. This latter interlock is necessary for reasons described under the subject of the "main motors."

The main field switches have already been described under the subject of the exciter switchboard; they are operated by the outboard small lever shown in the center of the main switchboard.

The speed levers are the two small inboard levers at the center of the main switchboard. They are connected by bell cranks and rods to the governor-control gear, as shown in Figure 2 and previously described under the subject of the "main turbine." These levers move on notched quadrants; they are moved from notch to notch when comparatively large changes of speed are quickly desired, but small changes of speed are made by vernier wheels. This type of lever has not proved very satisfactory in service, however, and a new type is being installed; this new type is similar to the one installed on the *Jupiter*, and consists of a worm and worm-wheel transmission which enables the operator to go through the full range of speed without lifting the lever from the quadrant; in addition, it will be possible to lift the worm out of engagement and move it in either direction when large changes in speed are quickly desired. The speed lever and field lever are mechanically interlocked so that the speed lever is automatically carried to the slow-speed position when the field switch is open. Also, the speed cannot be increased very greatly without carrying the field lever with it and increasing the excitation on the main field of the generator. This prevents dropping motors out of step by large increases of speed of the generator without corresponding increases in its excitation.

The steam limit levers are mounted directly underneath the field and speed levers and are similar to them in type, consisting of short arms moving on notched quadrants. The motion of the steam-limit levers is transmitted by rods and bell cranks to a stop on the operating gear of the main governor. This lever can be set to allow any desired number of valves to open on the main turbine and when once set the amount of power that can be developed by the turbine is limited by the amount of steam that the given number of valves will pass. If the load becomes such that the required amount of steam to maintain the speed is greater than that allowed by the steam limit, the turbine will simply slow down as would any other type of throttled engine. The relative position of the steam-limit stop and governor-operating mechanism is always shown at the main switchboard, as will be explained under the subject of electric interlocks. The necessity for the steam limit arises from the conditions that obtain when the ship makes a turn. In this case, the power required to maintain the revolutions the same as they were before the turn is enormously increased; since the turbine is at all times under the control of the governor, it will continue to maintain the same speed while the ship is turning that it did before the turn started, and this will result in overloading the main generator beyond its capacity; this will result in a drop in its voltage and the main motors will then drop out of step due to insufficient torque. But when the steam limit is set, the turbine will slow down before the excessive overload is reached on the generator, and the latter will then have sufficient voltage to furnish the necessary torque for holding the motors in step.

The D.C. interlocks, booster-field control, and main control wiring are shown in Figure 26. The booster is a motor generator set, the motor end of which is a constant-speed motor and the generator end of which has a separately excited reversible field. The field current for excitation of the main

generator passes in series through the booster generator and regulation of the strength of the main field current is obtained by varying the voltage of the booster generator. This either bucks or boosts the voltage of the 300-kw. generator, according to the throw of the field lever. The arrangement of the field windings and the booster-field rheostat is shown in Figure 26. The range of the booster generator is about 60 volts in either the buck or boost direction. Generally the boosting is only used when it is desired to put over-excitation on the main field of the generator during reversal.

The various indicating lamps and electric interlocks used in the operation of the machinery are shown in Figure 26. These are supplied by a double-throw transfer switch and a double-throw ordinary knife-blade switch. The transfer switch supplies current either from the exciter bus or from the after distribution room. In transferring from one to the other the circuit is not interrupted, but resistances are inserted directly in the two circuits so as to make it safe to transfer from one to the other without paralleling the generators. The supply bus from the exciter is fed through a double-throw switch which can be thrown on either exciter.

There is a balanced-relay in each main generator circuit which is shown at the top of Figure 26. This relay is normally in the open position and is held rigidly so by the solenoids in each circuit. In case of short-circuit on either of the phases of the generator itself, the motors, or the cable, the pull on the solenoids of this relay will become unbalanced, and the contacts in the D.C. circuit will be closed, tripping out the main field switch and taking all power off the generator in question.

Under-load relays are also provided for each generator circuit. The solenoids of these relays are adjustable and can be made to open at any desired value. Until this relay opens, the reversing and pole-changing levers are locked so they cannot be moved. A blue lamp indicates when this circuit is closed.

The main field switch, when closed, electrically locks the main generator disconnecting switches, motor disconnecting switches, and the doors to the cell structure of the main switchboard. This insures that none of these switches will be moved while the circuits are alive. There are red lamps for indicating when the switch is closed and green lamps for indicating when it is open. When only one generator, and consequently one field switch, is in use, the bus-tie switch will always be closed. The bus-tie switch is therefore provided with an auxiliary switch which connects the two interlocking systems on the two sides of the ship so that either field switch will then operate all interlocks.

White and red lamps are provided on each side of the board to show the relative position of the steam-limit stop and the main governor-control gear. The white light indicates an opening of about ⅛ inch (this is adjustable) between the stop and governor gear. When both lamps are lighted this indicates contact between the stop and governor gear, and when the white light goes out, leaving only the red lamp on, it indicates that the governor gear is hard up against the stop. This enables the operator to keep his steam limit properly adjusted for each speed of the ship.

The *New Mexico* is operated from low speeds up to 15 knots using one generator, and the motors connected for the 36-pole connection; from 15 knots to 17 knots one generator is used with the motors on the 24-pole connection; from 17 knots to full power, both generators are used and motors will be arranged for the 24-pole connection. There are thus three conditions of normal steaming and the operation of the ship for both ahead and astern motion will be described for each of these conditions. The operation is as follows:

(1) *Getting under way with one generator and motors on 36-pole connection.*—Assume that the maximum speed will not be over 15 knots. This means that one generator will be used and that the motors will be operated on the 36-pole con-

nection. Before reporting ready to get under way, the generator that it is desired to use would be tested out and then set to run at low speed; the bus-tie switch would be closed, and the generator-disconnecting switch would be closed in the low-voltage position. The field switch of the generator and the pole-changing and reversing levers would be open. On receiving a signal "Ahead,"

(a) The pole-changing switch would be thrown in the "36-pole" condition.

(b) The reversing lever would be thrown in the "Ahead" position.

(c) The field switch would be closed and brought up to the desired field strength.

(d) The turbine would then be brought up to the desired speed.

(2) *Reversing.*—With the ship going ahead under conditions described above, under (1), on receiving a signal "Astern,"

(a) The field switch would be opened and the speed lever brought to low speed at the same time.

(b) As soon as the under-current relay operates, the reversing levers would be thrown to the "Astern" position.

(c) The field switch would then be closed and over-excitation applied to the generator until the motors have reversed and come up to synchronous speed, when the field would be reduced to its normal value.

(d) The speed of the turbine would then be brought up to whatever is desired.

(3) *Getting under way with one generator and motors on 24-pole connection.*—In case it is desired to make the maximum speed greater than 15 knots, but less than 17½ knots, one generator will be used and the motors will be connected on the 24-pole condition. In this case, getting under way is exactly as described under (1), except that the pole-changing switches would be thrown into the 24-pole position.

(4) *Reversing.*—Assuming the ship to have gotten under way with conditions as described above, under (3), on receiving a signal "Astern,"

(a) Open the field switch and throw the speed lever to low speed at the same time.

(b) Throw pole-changers to 36-pole position.

(c) Throw reversing switches to "Astern" position.

(d) Apply over-excitation to the field of the main generator until motors have come into synchronism and then reduce field to normal value.

(e) Bring turbine up to desired speed.

(5) *Getting under way with two generators.*—In this case the bus-tie switch would be open and the two generator-disconnecting switches would be closed in the high-voltage position. Field switches would be open; generators would be running at slow speed and pole-changing and reversing switchs would be open. If signal "Ahead" is received, the operations carried out would be exactly as described under (3), except that it would be performed on two generators instead of one.

(6) *Reversing.*—After having gotten under way with conditions as above described under (5), if a signal "Astern" were received, the operations would be exactly as described under (4), except that they would be carried out for two generators instead of one.

SHAFTING AND BEARINGS.

There are four lines of shafting, each driven by one of the main propelling motors. Each line of shafting is composed of the following number of separate sections, all rigidly coupled together:

	Inboard shafts.	Outboard shafts.
Thrust shaft	1	1
Line shaft	2	1
Stern-tube shaft	1	1
Propeller shaft	1	1

Thrust Shafts.—The thrust shafts are fitted to thrust bearings of the adjustable horseshoe type, with steady bearings at each end for supporting the shafts. The shafts are hollow, with thrust collars and coupling discs forged integral with the shafts.

Line Shafts.—These shafts are hollow forged, with disc couplings at either end, and each is carried by two steady bearings.

Stern-Tube Shafts.—The stern-tube shafts are covered with a composition casing within the stern tubes and at the stern tube bearings. They are hollow forged, secured to the line shafts at inboard end by special disc couplings, and to the propeller shafts at outboard end by sleeve couplings. Two lignum vitæ lined bearings are provided for each stern tube shaft.

Propeller Shafts.—Each propeller shaft is carried by one lignum vitæ lined strut bearing. The shafts are hollow forged, taper turned at after end to suit the propeller hub, and composition bushed at the bearings.

Inboard Coupling.—Each inboard coupling consists of a sleeve secured to the stern-tube shaft by four keys, the end of the shaft being slightly upset for reception of the sleeve. Back of the sleeve is a collar made in halves and secured to the sleeve and to the coupling disc on the line shaft by fitted bolts. The shaft is prevented from pulling out of the sleeve when backing by the half collars, which shoulder against the upset on the end of the shaft.

Outboard Coupling.—Each outboard coupling is of the solid-sleeve type, taper bored for reception of shafts, and secured to each shaft by two feather keys and one cross key.

SHAFT DATA.

Thrust shafts, length, feet and inches	16–06
diameter, inches	15¼
at bearings, inches	15½
axial hole, inches	7⅝
collars, number	12

Thrust shafts, collars, thickness, inches	2
space between, inches	4
outside diameter, inches	23
inside diameter, inches	15½
Line shafts, forward, inboard, length, feet and inches	29–03
after, inboard, length, feet and inches	29–03
outboard, length, feet and inches	18–06
diameter, inches	15¼
at bearings	15½
axial hole, inches	7⅝
Stern-tube shafts, length, feet and inches	42–02⅝
diameter, inches	16⅞
axial hole, inches	8⅜
Propeller shafts, length, feet and inches	34–11¼
diameter, inches	16⅞
axial hole, inches	8⅜
Coupling discs, diameter, inches	30
thickness, inches	3½
Inboard coupling, diameter of sleeve outside, inches	30
inside, inches	16⅞
length of sleeve, inches	13
thickness of collar, inches	3½
Coupling bolts, number each coupling	8
diameter (taper) at face of coupling, inches	3¼
Outboard couplings, length of sleeve, inches	63¼
diameter of sleeve, inches	22½

BEARING DATA.

Thrust bearings (white-metal lined bearings and shoes):	
Steady bearings, number each	2
diameter, inches	15½
length, inches	19
Thrust shoes, number each	12
effective surface, square inches, ahead	1,782
astern	1,635.5
Line-shaft bearings (white-metal lined):	
diameter, inches	15½
length, inches	26
Stern-tube bearings (lignum vitæ lined):	
forward bearing, diameter, inches	19⅛
length, inches	48
after bearings, diameter, inches	19⅛
length, inches	60
Strut bearings (lignum vitæ lined):	
diameter, inches	19⅛
length, inches	72

TURNING GEAR.

Each line of main shafting is provided with a motor-driven, reversible turning gear. The motors are 5-horsepower each. Each gear is so arranged that it can be operated by hand, a long ratchet wrench being provided for that purpose.

PROPELLERS.

There are four three-bladed propellers, with the blades cast solid with the hubs. The starboard propellers are right-hand and the port propellers are left-hand; all propellers turn outboard when driving the vessel ahead. They are of manganese bronze and the hubs are taper bored to fit the shafts, to which they are secured by a key and locked nut on the end of each shaft. The blades are machined true to pitch, and polished all over, including the hubs.

PROPELLER DATA.

Diameter of propeller, feet and inches	13–5
hub, feet and inches	2–9
Pitch, feet and inches	15–2
Ratio of diameter to pitch	0.8846
Area, projected, square feet	50.43
helicoidal, square feet	61.22
disc, square feet	141.38
Ratio, projected to disc area	0.3567
helicoidal to disc area	0.4333
Heighth of lower tip of blade above keel, inches, inboard	11⅛
outboard	20⅛
Immersion of upper tip of blade, feet and inches, inboard	15–07⅞
outboard	14–10⅞

MAIN CONDENSING APPARATUS.

Main Condensers.—There are two main condensers, one for each main turbine. They are installed in the same compartments with the main turbo-generators and supported by substantial foundations above the turbines. The tubes are straight, rolled into the tube sheets at one end and gland packed at the other end. The condensers are of cylindrical

cross-section and conforms to Navy Standard practice. The principal dimensions follow:

Inside diameter, feet and inches	10–3
Thickness of shell (steel), inch	0½
Length between tube sheets, feet and inches	10–6
Thickness of tube sheets, inches	1⅜
Tubes number	8,901
diameter, outside, inch	0⅝
thickness, inch	0.065
Cooling surface, square feet	15,300
Size of main exhaust nozzles, inches	56½×75
Diameter of air-pump suction, inches	10
condensate pump suction, inches	6
auxiliary exhaust nozzle, inches	10
circulating-water inlet and outlet, inches	26

Dry Air Pumps.—A vertical, motor-driven, direct-connected, double-acting, single, dry-vacuum pump is provided for each main condenser. They are located in the center engine room and are arranged to draw air and non-condensible vapors from the condensers and discharge same to the atmosphere above the main deck. The pump suctions are cross-connected so that either pump may be used on both condensers in emergencies. The pumps and motors are of the following principal characteristics:

Make of pump	Wheeler Condenser & Engineering Co.
Diameter air cylinder, inches	30
Stroke, inches	18
R.p.m.	100
Diameter suction, inches	10
discharge, inches	6
Horsepower of motor	40
Make of motor	G. E. Co.

Condensate or Hotwell Pumps.—The condensate is drawn off the bottom of each main condenser by a horizontal motor-driven, direct-connected, double-inlet centrifugal pump discharging into the filter chambers of the feed and filter tanks. The pumps are located in the center engine room and have a cross-connection in their suctions so that either may be used on both main condensers, as is the case with the air pumps.

The pumps and motors are of the following principal characteristics:

Make of pump	Alberger Pump & Condenser Co.
Capacity, gallons per minute	500
Diameter impeller, inches	9½
suction, inches	8
discharge, inches	6
Horsepower of motor	20
R.p.m.	1,800
Make of motor	G. E. Co.

Main Circulating Pumps.—Circulating water for each main condenser is supplied by a horizontal motor-driven, direct-connected, double-inlet, centrifugal pump located in the center engine room. They are also arranged for pumping on the engine-room bilges in emergencies. The pumps and motors are of the following principal characteristics:

Make of pump	Alberger Pump & Condenser Co.
Capacity, gallons per minute	21,200
Diameter, impeller, inches	26
sea suction, inches	26
bilge suction, inches	21
discharge (at pump), inches	24
Horsepower of motor	250
R.p.m.	500
Make of motor	G. E. Co.

FORCED-LUBRICATION SYSTEM.

The main turbo-generator sets, the main propelling motors and the thrust bearings are provided with forced lubrication.

The pumping equipment, located in the center engine room, consists of four motor-driven, geared, oil pumps of the Kinney type, four Schutte-Koerting, type Z, oil coolers (two No. 5, of 127 square feet of cooling surface each, and two No. 7, of 212 square feet of cooling surface each), and two 7-inch by 8-inch by 12-inch vertical, piston, double-acting, single, Blake pumps for supplying cooling water to the oil coolers. Two oil drain tanks of 1,000 gallons' capacity each and four discharge strainers are also provided.

The lubricating-oil pumps draw oil from the drain tanks and discharge same through pressure reducing bafflers to the oil coolers, thence, via the lubrication main and its branches, to the bearings. The oil coolers can be by-passed if so desired. The oil from the bearings is led back to the drain tanks through drain lines, thus completing the cycle, which is indefinitely repeated. Cut-out valves are provided as necessary for isolating units not in operation.

A thermometer is fitted at each oil-cooler inlet and outlet, and also at each bearing.

Oil for the main turbine-governor gear is also supplied by the lubricating-oil pumps at a pressure of about 80 pounds per square inch. In order to keep this excessive pressure off the oil coolers and lubrication lines, bafflers are placed in the pump discharges beyond the branches to the governor gear, which reduce the pressure on the lubricating system to about 40 pounds per square inch.

Reserve lubricating oil is carried in two storage tanks of 3,000 gallons combined capacity. These tanks are provided with deck-filling connections and are arranged for readily replenishing the forced-lubrication system.

Two 1,000-gallon lubricating-oil treating tanks, fitted with steam heating coils, are also provided. The forced-lubrication pumps are arranged to pump the oil from the system into these tanks when it requires cleansing.

WATER SERVICE SYSTEM.

The water service for the main turbines, thrust and spring bearings and the stern tubes is supplied from connections taken off the main circulating pumps, fire and bilge pumps and oil-cooler circulating pumps discharges. There is also an emergency connection from the fire main.

Branches, controlled by separate valves or cocks, are led to the various bearings and auxiliaries requiring cooling. Hose valves are also provided at convenient locations.

The drains from the main turbine bearings discharge into funnels so that they can be readily observed. The funnels are piped to a small drain tank in the center engine room, which has connections to the fire and bilge-pump suctions for pumping overboard. All other drains are closed circuits, uniting in a common main, and discharge overboard through the auxiliary overboard discharge sea chest on the center engine room.

FIRE AND BILGE PUMPS.

There are seven fire and bilge pumps, four located in the center engine room and one in each fireroom.

They have suctions from the sea and the drainage systems, and discharge to the fire and flushing mains and overboard. The engine-room pumps also have a suction from the water-service drain tank and a discharge to the oil coolers and water service.

The pumps are of the following characteristics:

Type—Worthington, vertical, piston, double-acting, single.
Size—12 inches by 10 inches by 18 inches.
Diameter of suction and discharge, 5 inches.

BOILERS.

There are nine Babcock & Wilcox water-tube boilers, arranged three abreast in three separate watertight compartments.

The boilers have a total heating surface of 55,458 square feet, exclusive of superheating surface, and are designed to operate the entire machinery plant at full power, with an average air pressure in the ash pits of not more than 6 inches of water. They are equipped for burning fuel oil.

Uptakes of the usual design are provided, all uniting into a common smoke pipe at the main deck. The smoke pipe is 14 feet 3 inches inside diameter, and 92 feet 6½ inches high above the bottom of the furnaces.

BOILER DATA.

Number	9
Pressure, working, pounds per square inch	280
test, pounds per square inch.....water, 420; steam	330
Height to top, external, feet and inches	15 1 11/16
Length on floor, feet and inches	9–0¼
Width on floor, feet and inches	17–3⅝
Drum, diameter, inside, inches	42
length, feet and inches	18–0¼
thickness, inch	11/16
Number of tube headers, each boiler	56
Distance between headers, feet and inches	9–00
Number of tubes, each boiler:	
2-inch generating tubes, 134 mils. thick	1,210
4-inch generating tubes, 165 mils. thick	28
2-inch superheater tubes, wing boilers, 134 mils. thick	46
2-inch superheater tubes, wing boilers, 165 mils. thick	46
2-inch superheater tubes, center boilers, 134 mils. thick	54
2-inch superheater tubes, center boilers, 165 mils. thick	54
Number of furnaces, each boiler	1
Furnace volume, each boiler, cubic feet	567
Heating surface, each boiler, square feet:	
Generating tubes	6,162
Superheater tubes, wing boilers	470
center boilers	552
Area through smoke pipe, square feet	149.5
Area through smoke pipe / Furnace volume	0.0293
Kind of forced draft	Closed firerooms.
Atomizers, number each boiler	7
type	Peabody mechanical atomizers.
Diameter of main steam stop valve, inches	7
main feed stop and check valves, inches	3½
auxiliary feed stop and check valve, inches	3
surface-blow valves, inches	01½
bottom-blow valves (two), inches	01½
safety valve (two), duplex, inches	04

FUEL—OIL SYSTEM.

The fuel-oil tanks are filled through eight 6-inch connections, two on each side of the ship forward and two on each side of the ship aft.

The two starboard forward filling connections are combined into an 8-inch pipe, which is led to a relay tank located on the third deck, starboard side. The port forward filling pipes are similarly connected to the same relay tank. There is another relay tank aft, located on the third deck, port side; the after filling connections are similarly led to this relay tank.

From each relay tank two 8-inch pipes are led to the distributing manifolds in the center engine room and forward fireroom.

In addition to the main filling connection described above, there are four emergency filling connections, two starboard and two port, each connecting to one of the four distributing manifolds in firerooms Nos. 1 and 3.

From the bottom of each fuel oil tank 4½-inch pipes, as required, are led to the various distributing manifolds, which are inter-connected so that any pump may be used on any tank. Each tank is fitted with a high suction and a low suction.

Two of the oil tanks, just forward of fireroom No. 1, are used as service tanks; similarly two others, just aft of fireroom No. 3. The fuel oil for the burners is normally taken from these tanks, which are replenished from the storage tanks as described elsewhere, but the system is so arranged that oil may be used from any storage tank direct.

Two 7-inch by 8-inch by 12-inch Blake, vertical, piston, double-acting, single, light-service booster pumps are installed in each fireroom Nos. 1 and 3. The speed of each pump is controlled by a Foster pressure governor. Each pump has a 4½-inch suction connection from the adjacent distributing manifold, and a 4-inch discharge to the distributing manifold, adjacent emergency deck-filling connection, fuel-oil main and overboard; the latter connection is portable. The booster pumps are also arranged to pump from the storage tanks and discharge to the service tanks while the service pumps are pumping from the service tanks. In addition to the connections noted above, the booster pumps also have a discharge to

the deck for fueling the launches and motor boats; also a branch to the galleys and foundry.

There are two heavy-pressure, horizontal, turbo-driven, rotary service pumps of the Kinney type in each fireroom. Mason pressure governors are provided to control the speed of the units; the turbines are also provided with centrifugal, emergency-trip governors to prevent overspeeding. They have 3-inch suctions from the fuel-oil main and service tanks, and each discharge, through an oil meter and an oil heater, to the atomizer supply pipes. Small by-passes are provided between the discharge and suction sides of the pumps for use when only a small amount of oil is being burned.

The oil meters are of the Bowser type, and are arranged so that they can be by-passed.

The oil heaters are of the Griscom-Russell type, each of sufficient capacity to heat 14,000 pounds of oil per hour from 70 degrees to 240 degrees F., with steam at 250 pounds' pressure. The oil heaters are arranged so that they can be by-passed.

The atomizers, seven per boiler, are carried in one row on the boiler fronts, with stop cocks at each burner for individual control. They are of the Peabody mechanically atomized type. The atomizer-supply pipe to each boiler is fitted with a cutout valve.

Steam coils are fitted near the ends of all suction pipes in all tanks; steam connections for boiling-out the tanks are also provided.

For raising steam with no air pressure available, a 1½-inch by 3½-inch duplex hand pump is fitted in each fireroom, capable of supplying oil at 200 pounds' pressure to two atomizers.

PNEUMERCATOR SYSTEM.

For ascertaining at any time the amount of fuel oil in the various double-bottom tanks, the Parks pneumercator system is installed.

The installation consists of a small semi-spherical balance chamber placed at a predetermined location near the bottom of each tank. The interior of the balance chamber is in communication with the tank, through a hole in the side of the chamber, and a small pipe connects the top of the chamber with its recording instrument in the engine rooms and fire-rooms at the fuel-oil manifolds. The recording device consists of a glass mercurial tube provided with scale, calibrated to suit the tank to which it is connected.

The instrument operates on the following principle: The pressure due to the head of oil in the tank compresses the air in the chamber and connecting pipe line, which in turn causes the mercury to rise or fall in the tube in direct ratio to the pressure exerted. The tank contents is read off on the scale, as in the case of an ordinary thermometer.

In order to guard against overflowing the tanks when taking aboard fuel oil, an annunciator is installed in connection with the recording device for indicating and signalling when any tank is 95 per cent. full.

FORCED DRAFT BLOWERS.

Forced-Draft Blowers.—Each fireroom has three forced-draft blowers. They are located in specially constructed blower rooms just below the third deck and above the working flat in front of the boilers.

The fans are 17½ inches outside diameter, Keith, horizontal, double-inlet type, each capable of delivering 23,000 cubic feet of air per minute against a static pressure of 8 inches of water, when running at 1,800 revolutions per minute.

Each fan is direct-connected, by a flexible coupling, to a horizontal Terry steam turbine.

Air is supplied the blowers from a large air duct on the second deck.

The blowers discharge the air into the firerooms through short trunks, extending through openings in the blower-room floor.

Deflectors are fitted at the trunk outlets, so arranged that the air will be properly diffused throughout the firerooms.

An automatic hinged shutter, closing by back pressure, is fitted in each blower discharge to prevent the loss of air pressure in the fireroom in case a blower stops.

MAIN STEAM PIPING.

The main steam piping is arranged in two symmetrical systems, one on each side of the vessel. The two lines are cross-connected in each fireroom. The branches from the boilers are 7 inches in diameter each, and the lines proper are 8½ inches in the forward fireroom, increasing to 11, 12 and 14 inches at each successive boiler connection, the latter size continuing to the main engine's throttle valves. The piping is seamless-drawn steel.

AUXILIARY STEAM PIPING.

The auxiliary steam pipe is seamless-drawn steel.

From the main steam pipe in the after fireroom there are two 6½-inch connections leading aft through the after dynamo room and forming a loop in the center engine room. The engine room auxiliaries take their steam from this loop.

In the firerooms the auxiliary steam piping consists of branches from the main steam piping with sub-branches to the various auxiliaries.

The ship's turbo-generator sets take their steam off the main steam cross-connection in the firerooms Nos. 1 and 3.

Stop valves are fitted in all branches and sub-branches.

AUXILIARY EXHAUST PIPING.

A single copper auxiliary exhaust pipe, ranging in size from 10 inches to 14 inches, is led throughout the machinery spaces with branches leading elsewhere as required for the various auxiliaries. Connections are provided to direct the exhaust steam into either main condenser, either dynamo condenser,

either feed-water heater, evaporators, or into the atmosphere through the escape pipe at will.

Stop valves are fitted in all branches at the main.

ESCAPE PIPES AND ATMOSPHERIC EXHAUST.

There are two 15½-inch escape pipes, one forward and the other aft of the smoke pipe. Branches from these pipes are led through the third deck to each fireroom, where they are further subdivided and connected to the individual safety valves through 6-inch branches.

There is a 10-inch branch in firerooms Nos. 1 and 3 from the auxiliary exhaust to the escape pipe for atmospheric exhause. A stop valve is fitted at each connection to the escape pipe.

There is also a 10-inch atmospheric exhaust connection, fitted with automatic valve, from each dynamo turbine to the escape pipes.

All the above piping is copper.

MAIN AND AUXILIARY FEED SYSTEMS.

Feed and Filter Tanks.—Two feed and filter tanks of 5,200 gallons' capacity each, are located in each center engine room. A filter chamber of about 900 gallons capacity is in the top of each tank. Each filter chamber has an inner bottom of loose perforated plates and is divided into three compartments, in which is placed the filtering material, by vertical division plates. These partitions are so arranged that the water in passing through the filter will flow under and over in succession, thus assuring the filtering material being always submerged.

Each tank is provided with the following principal connections:

1 6-inch main hotwell pump discharge;

1 3-inch dynamo condenser hotwell pump discharge;

1 4-inch reserve-feed pump discharge;

1 10-inch cross-connecting pipe;
1 8-inch overflow;
steam drain discharges;
vapor pipes.

Feed-Water Heaters.—There is a Reilly multicoil feed-water heater in each fireroom.

Each heater has 239.85 square feet of heating surface, and a rated capacity of 166,700 pounds of water per hour from 100 degrees F. to 200 degrees F., with steam at 10 pounds gage.

The heating agent is the exhaust steam, a back pressure of about 10 pounds gage being kept in the auxiliary exhaust line for this purpose by means of a spring-relief valve at each connection to a condenser, opening toward the condensers.

Main Feed Pumps.—There are three horizontal, direct-connected, turbine-driven, centrifugal main feed pumps located in the center engine room. The speed of the pumps is controlled by "Ideal" pressure governors. Centrifugal, emergency-trip, over-speed, governors are also provided.

The following is the principal pump and turbine data:

Pump—Make and type, Cameron, horizontal, 4-stage.
Capacity, each, 500 G.P.M.
Head, 900 feet.
Turbine—Make and type, G. E. Co., horizontal, single stage.
Horsepower each, 190.
R.p.m., 3,600.

Auxiliary Feed Pumps.—An auxiliary feed pump is installed in each of the three firerooms. They are of the Blake, 16-inch by 10½-inch by 24-inch, vertical, piston, double-acting, single type. "Foster" pressure governors are provided for the speed control of these pumps.

Reserve Feed Pump.—A Blake, 7-inch by 8-inch by 12-inch, vertical, piston, double-acting, single, reserve feed pump is installed in the center engine room. It is arranged to pump water from the reserve-feed tanks to the main feed tanks. It also has a suction connection from the hotwell-pumps cross-connection.

Piping Arrangement.—The feed piping is seamless-drawn copper.

Each main feed pump has a 6-inch independent suction from the main feed tank cross-connecting pipe.

The main feed pumps discharge via the feed-water heaters, or by-pass same if desired, to the boilers. The discharge from each pump is 5 inches, uniting in the center engine room into an 8½-inch connection, leading forward and diminishing in size as it advances to 5 inches in the forward fireroom. The connections to the feed-water heaters are 5 inches in diameter each, and the supply branches to each boiler are 3½ inches in diameter.

The auxiliary-feed suction main is taken off the feed tank's cross-connecting pipe. This main is 10 inches in diameter, reducing in size as it leads forward to the auxiliary feed pumps. Each pump has a 6-inch suction connection.

Each auxiliary feed pump is arranged to discharge to the boilers in its compartment, through the feed-water heater or direct. They can also discharge into the main feed-discharge line to the boilers in other compartments. The discharge connection from each pump is 5 inches in diameter, and the supply branches to the boilers are 3 inches diameter each.

INTERIOR COMMUNICATION.

The customary engine and fireroom telegraphs, gongs, telephones, voice tubes, etc., are fitted for transmitting orders and signaling to the various machinery compartments and other parts of the vessel.

AIR COMPRESSOR PLANT.

An air compressing plant is provided for ejecting gas from the guns, cleaning boiler tubes, cleaning dynamos, running pneumatic tools, etc.

The plant is divided into two parts, each consisting of two motor-driven, geared, compressors and three air-storage tanks,

together with switchboards and appurtenances. One group of equipment is located forward on the third deck in the windlass room, and the other is located on the third deck aft of No. 3 barbette and enclosed in a wire-mesh bulkhead.

The air compressors are automatically controlled so as to maintain a constant pressure on the system. They are also provided with automatic unloading valves so that after shutting down the compressor will always be started up in the unloaded condition.

There is a pneumatic main running through the machinery space with nozzles in each compartment. Branches from this main are led to the general workshop, dynamo, ice-machine and evaporator rooms, galleys, etc., as required. The main is connected to all the air compressors.

The main for ejecting gas from the guns is independent of the above mentioned pneumatic main, and is likewise connected to all the air compressors.

The following are the principal characteristics of the air compressors:

Make—Worthington, motor-driven, geared.
Size—15-inch by 12¾-inch by 7-inch stroke.
Capacity, each—175 cubic feet of free air per minute at 150 pounds' pressure per square inch.
R.p.m., compressor, 300.
R.p.m., motor, 1,600.
Horsepower of motor, 45.
Capacity air-storage tanks, each, cubic feet, 34.

EVAPORATING AND DISTILLING APPARATUS.

The evaporator room is on the second platform between the engine and firerooms, starboard side. The plant is arranged to operate in double-effect with live steam at 70 pounds gage in the first-effect steam coils, or in single-effect when using auxiliary exhaust steam at about 10 pounds gage. Under the latter condition a slight vacuum is maintained on the distillers, an air pump being provided for the purpose.

The plant has a combined nominal capacity of 40,200 gallons of water per 24 hours, when operating in double effect, with a 40 per cent. overload capacity when clean.

EVAPORATOR DATA.

Type	Schutte-Koerting, vertical U-tubes.
Number	6
Diameter, inside, inches	38⅛
Height over all, feet and inches	7–6
Tubes, number (each)	54
diameter of pipe, inches	01¼
thickness of pipe, inch	0.083
Heating surface, square feet (each)	113
Diameter of steam connection, inches	02½
vapor nozzle, inches	07
feed valve, inches	01¼
blow valve, inches	02½

DISTILLER DATA.

Type	Bureau, straight tube.
Number	4
Diameter, inside, inches	22
Length, over all, feet and inches	6–5
Tubes, number (each)	327
diameter, inch	00⅝
thickness, inch	00.065
Cooling surface, square feet (each)	240
Diameter of circulating water inlet and outlet, inches	06
vapor inlet, inches	07
drain, inches	03

EVAPORATOR VAPOR FEED WATER HEATER DATA.

Type	Bureau, U-tube.
Number	2
Tubes, number (each)	12
diameter, inside inches	02½
thickness, inch	0.083
Heating surface, square feet (each)	24.9
Diameter of feed inlet and outlet, inches	03
vapor inlet and outlet inches	07

EVAPORATOR COIL DRAIN FEED WATER HEATER DATA.

Type	New York Navy Yard, U-tube.
Number	2
Tubes, number (each)	6
diameter (I.P.S.), inches	1

Tubes, thickness, inch..... 0.134
Heating surface, square feet (each)..... 12.5
Diameter of feed inlet and outlet, inches..... 1¼
coil drain inlet and outlet, inches..... 1½

PUMP DATA.

Evaporator feed pumps:
Number and type—(2)—Worthington, vertical, piston, double-acting, single.
Size.....4½-inch by 5-inch by 6-inch.
Evaporator brine pump:
Type.....Worthington, vertical, piston, double-acting, single.
Size.....4½-inch by 5-inch by 6-inch.
Distiller circulating pumps:
Number and type—(2)—Worthington, horizontal, turbine-driven, centrifugal.
Capacity (each), G.p.m. at 25 feet head..... 800
Turbine, type .....Terry.
R.p.m. 2,000
Horsepower (each) 12
Distiller fresh-water pumps:
Number and type—(2)—Worthington, vertical, piston, double-acting, single.
Size.....4½-inch by 5-inch by 6-inch.
Distiller air pump:
Type—Worthington, vertical, piston, double-acting, single, feather-weight.
Size.....6-inch by 10-inch by 8-inch.

The quantity of fresh water produced by the plant is measured by a Cochrane precision meter.

GENERAL WORKSHOP.

The following machine tools, each driven by its own electric motor, are installed in the machine shop on the third deck, over the center engine room:

No.	Description.	Make and H.P. of motor.
1	28-inch by 48-inch swing, extension-gap lathe.	
1	16½-inch swing lathe; American Tool Works	Reliance, 3 H.P.
1	14-inch swing lathe; American Tool Works.	Reliance, 3 H.P.

1 8-inch swing precision lathe.
1 24-inch column shaper; American Tool Works ..... Reliance, 3 H.P.
1 30-inch radial drill; American Tool Works. Reliance, 3 H.P.
1 16-inch sensitive drill, Sipp Machine Co... ¾ H. P.
1 Universal milling machine; Kearney & Trecker ..... Reliance, 3 H.P.
1 Floor grinder; Willey ..... Reliance, 2 H.P.
2 Portable cylinder-boring machines;
1 Portable surface grinder; Cincinnati Electric Tool Co. ..... 1 H.P.
1 4-inch hacksaw.

BLACKSMITH SHOP.

A blacksmith shop is provided on the superstructure deck, in the foundry enclosure. It is equipped with one portable and one permanent forge, together with anvil and all necessary tools and fittings. The permanent forge is oil burning; it is fitted with an electrically-driven blast fan. The portable forge is of the coal-burning folding type.

FOUNDRY.

A small foundry is installed on the superstructure deck. The outfit consists of a small marine, oil-burning, tilting, crucible furnace, together with adequate allowance of crucibles and the necessary apparatus for handling and pouring the metal.

PIPE AND COPPERSMITH SHOP.

The pipe and coppersmiths' equipment is located on the third deck in the forward boiler hatch, and consists of the following:

1 Motor-driven, pipe-cutting and threading machine; 2-inch to 6-inch capacity;
1 tinner's combined punch and shears;
1 tinner's slip roll forming machine;

1 tinner's beading machine;
1 tinner's cornice brake;
1 oxyacetylene welding outfit;
1 oil forge;
vises, work-benches, etc.

ELECTRIC PLANT.

There are two dynamo rooms, one just forward of the forward boiler room and the other just aft of the after boiler room. They are on the second platform level.

The distribution rooms, two in number, are located on the first platform, one over each dynamo room. They contain the lighting, power, searchlight and generator boards combined.

The generator installation consists of four horizontal, 6-pole, compound-wound, 300-kilowatt, 3-wire, General Electric generators, two in each dynamo room; they are gear-driven by three-stage horizontal Curtis turbines, designed to run at 5,037 revolutions per minute, with steam at 200 pounds gage. Each generator will deliver at normal load 1,250 ampères of current at 240 volts, when running at 1,000 revolutions per minute. The generators are capable of delivering one-third overload for two hours without injury.

Steam for the generators is taken off the main steam cross-connection in the forward and after firerooms, for the forward and after generators, respectively. The branch to each generator is 4 inches and fitted with stop, reducing and throttle valves.

The generators are arranged with condensing and atmospheric exhaust, the latter connections, via escape pipe, are 10 inches and fitted with automatic valves permitting instantaneous change over in emergency.

Each pair of generators has an independent condensing apparatus, located below the generators in a separate room in the hold, with access hatch between the two levels. Each condensing apparatus consists of one condenser and vacuum

augmenter, together with independent air pump, circulating pump, hotwell-tank pump and hotwell tank of 300 gallons' capacity.

The dynamo condensers also handle the exhaust in port.

DYNAMO CONDENSER DATA.

Number	2
Inside diameter, feet and inches	4–05
Thickness of shell (steel), inch	0⅜
Length between tube sheets, feet and inches	8–6⅛
Thickness of tube sheets, inch	01⅛
Tubes, number (each)	1,112
diameter, outside, inch	00⅝
thickness, inch	00.065
Cooling surface, square feet (each)	2,000
Diameter of exhaust nozzles, generators (2), inches	18
augmenter suction, inches	05
air-pump suctions, inches	07
circulating-water inlet and outlet, inches	09

AUGMENTER CONDENSER DATA.

Number	2
Inside diameter, inches	12½
Thickness of shell (composition), inch	0⅜
Length between tube sheets, inches	32
Thickness of tube sheets, inch	01
Tubes, number (each)	116
diameter, outside, inch	00⅝
thickness, inch	00.049
Cooling surface, square feet (each)	50.61
Diameter vapor inlet, inches	04
air-pump suction, inches	04
circulating-water inlet and outlet, inches	08

PUMP DATA.

Dynamo condenser circulating pumps:

Number and type—(2)—Worthington, horizontal, turbine-driven, centrifugal.	
Capacity (each), G.p.m. at 20 feet head	2,000
Turbine, type	Terry
R.p.m.	1,300
Horsepower (each)	15.7

Dynamo condenser air pumps:

Number and type—(2)—Worthington, vertical, twin, beam, single-acting.

Size.—18-inch by 18-inch by 18-inch by 12-inch.

Dynamo condenser hotwell pumps:

Number and type—(2)—Worthington, vertical, piston, double-acting, single.

Size.—4½-inch by 6-inch by 6-inch.

TRIALS.

The *New Mexico* has had two sets of official trials. The first trials were conducted after the ship had been out of dry dock about seven weeks, and this increased the shaft horsepower necessary to make the guaranteed speed of 21 knots by over 4,000 horsepower, so the ship was docked, cleaned and painted, and a second set of trials were run when the horsepower came down to 29,100 for 21 knots. The ship averaged 21.08 knots for her four-hour full-power run and 21.31 knots was the average of the five high points on standardization.

The turbine water rates per shaft horsepower were considerably better on the first trials than on the second, owing to the fact that the first trials were run in smooth water, while the second trials were run in rough weather, which increased the horsepower for the speed over that obtained on the standardization runs and for which no allowance was made, as the standardization horsepower was used. The water-rates obtained on the trials were as follows:

Speed.	Water rate first trial.	Water rate second trial.
Full speed	12.29	12.01
19 Knots	11.926	12.33
15 Knots	11.667	12.475
10 Knots	14.223	13.96

These water rates include the steam from the 300-kw. exciters, which furnished power for excitation and also for driving the main air pumps, main circulating pumps, main condensate pumps, forced-lubrication pumps, and ventilating blowers for the main motors.

In addition to the trials for steam and fuel consumption, trials were made to demonstrate satisfactory backing qualities when steaming at full power; also to determine satisfactory performance when turning with full rudder and at full speed, both in the case of propulsion by one generator and also by two generators. Numerous other trials were conducted, such as standardizing with two screws driving and two running idle, and considerable new data was obtained in regard to the action of screw propellers. In all the tests carried out the machinery gave very satisfactory performance.

The average trial data obtained on the second trial is given in the following table:

	4-hour full power trial.	4-hour 19-knot trial.	4-hour 15-knot trial.	1-hour 10-knot trial.
Steam at boilers, lbs. gage	278.6	274.8	273.8	
turbines, lbs. gage	272.1	274.2	274.6	277.5
turb. 1st stage, lbs. gage.	139.7	104.4	86.4	42.5
Vacuum, inches	29	29.5	29.8	30.
Barometer, inches	30.83	30.77	30.92	30.79
Fireroom air pressure, inches of water	4.1	3.4	2.3	1.7
Feed water temperature, deg. F.	182.8	189.7	203.4	208.9
Main generators, volts	4257.	3740.	2915.6	1950.
ampères	1873.5	1565.	2206.2	1600.
field, volts	171.7	152.15	143.8	118.
field, ampères	318.25	285.6	290.	245.
r.p.m.	2042.	1825.5	2012.5	1450.
Main motors, ampères	994.5	860.3	572.5	417.5
r.p.m.	167.69	152.2	115.35	80.49
Slip of propellers, per cent.	16.	14.97	13.24	14.83
Speed in knots	21.08	19.37	14.98	10.26
S.H.P. (from curve)	31197.	23233.	9648.	3690.

BALANCING MACHINES.

F. G. HECHLER, ASSOCIATE.*

The various articles on balancing which have recently appeared in this and other journals attest to the increasing interest manifested in this very important subject. The use of high speed rotative machinery has brought the matter prominently before the marine engineer, and he sees in it a possible solution for some of the baffling problems of excessive vibration that confront him. A little reflection will make it clear that the supports for machinery on shipboard are in general less rigid than in stationary work, and that the possibility of setting up a vibration of the ship's frame which will be in synchronism with that of a rotating machine, is by no means remote. It is not unusual to find that at certain speeds, which must be maintained when cruising in squadron formation, the vibration is very trying on the personnel. This phenomenon of vibrational "Critical Speed," as it is often called, is in many cases probably due to synchronism between an unbalanced turbine rotor or propeller, for instance, and the hull. In addition to the discomfort of the crew, the unbalance causes undue wear of the bearings, commutation and lubrication troubles and a more or less serious loss of power.

Although the lack of proper balance is quite generally recognized, there is considerable confusion regarding the subject and especially as to the methods to be used in securing both static and dynamic balance. Only recently a midshipman of the first class, who in a few months will become an officer, said to the writer that he had been told that the subject was so difficult that only two or three people knew the formulae therefor, and that they could get almost any price for their services! It is to be hoped that such ideas of the subject may soon be dispelled.

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In the past, the main reliance has been placed on securing satisfactory static balance by using the ordinary parallel ways, or sometimes roller or ball bearings. Except under the most favorable conditions of light weight and small shaft, combined with a relatively large radius of gyration, the static balance thus obtained will not be very exact as the method is not sufficiently sensitive. However, if after the static balance is satisfactorily adjusted, and the machine runs smoothly at all speeds, well and good; if not, an effort is made, usually by trial and error, to improve conditions by adding weights here and there until by chance the running becomes satisfactory. At best the method is slow and uncertain.

Occasionally, when the importance of the subject was more fully appreciated, balancing machines were designed and built. These machines have usually been called "dynamic balancing machines," but have in reality been a rather curious mixture of both static and dynamic, in that no provision was made for properly separating the two effects. Unless a body is first put in perfect static balance it is useless to attempt to balance it dynamically, for the results will surely be misleading and may be disastrous. These machines have depended usually either on the use of a floating bearing at each end, with all motion suppressed except in the horizontal plane, or on the one-point-suspension principle, as for instance, when a flywheel, pivoted on a cone near its center of gravity, is rotated in a horizontal plane. In both cases, the body runs out and the "heavy spot" is marked in some way. Usually the body is rotated in both directions, giving two distinct sets of marks from which the plane of unbalance must be determined. Then too, strange though it may seem, the effort was often made to correct what was assumed to be a dynamic unbalance by adding or removing weight at one point only! It can readily be shown that neither the floating bearing nor the one-point-suspension method is based on sound theoretical grounds, and hence machines built in accordance therewith are not, as a rule, susceptible of satisfactory use. However, in the hands of

capable and experienced men some very good operating results have apparently been obtained, but not with the definite directness and ease that is possible.

Credit for developing and applying the correct methods based on a rational treatment of the subject is due Mr. N. W. Akimoff. In a paper presented at the New Orleans meeting of the American Society of Mechanical Engineers in April, 1916, Mr. Akimoff first presented the subject in a most interesting and readable manner. Since then, however, great progress has been made in the application of the fundamental principles there laid down.

The subject is of great practical importance, and a description of the design and method of operation of the two leading makes of balancing machines now in the field will be of interest. Both the balancing machines to be described have been under test at the U. S. Naval Engineering Experiment Station, Annapolis, Maryland, within the past year.

Before describing the machines, however, it is desirable to examine briefly the meaning of static and dynamic balance. Fig. 1 represents a body symmetrical about the axis, but which

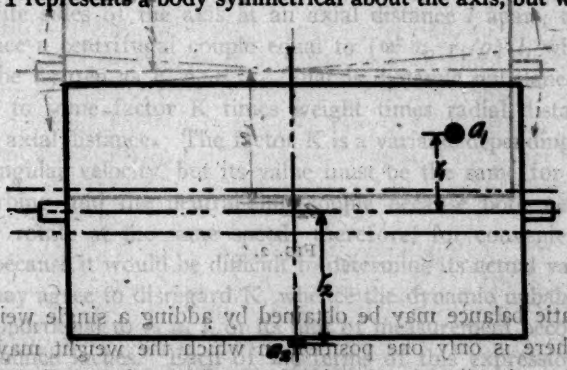


FIG. 1.

is out of static balance because of the "heavy spot" a_1 at a radius r_1. If such a body is placed on sensitive parallel ways it will always come to rest with a_1 vertically below the axis.

When the body is rotated, the unbalanced force due to a_1 will tend to make the axis "throw out," as indicated by the dotted center lines; that is, the body suffers a translation. To establish static balance it is only necessary to add one weight as a_2 at radius r_2 in the same axial plane as a_1, and such that $a_2 r_2 = a_1 r_1$. In general the "heavy" spot is not due to a single disturbing element, but represents the resultant effect of all unbalanced elements, and hence static balance for the entire body may be obtained by adding or removing a single weight. Evidently a body is in static balance when its center of gravity lies on the axis of rotation.

Dynamic unbalance in a statically balanced body is caused by a centrifugal couple in some axial plane. The centrifugal couple tends to produce rotation about an axis through the center of gravity of the body as shown by the arrows in Fig. 2.

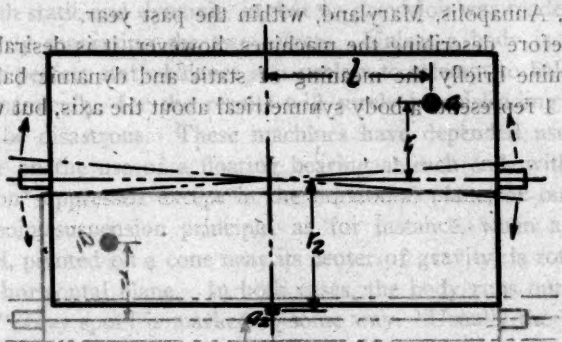


FIG. 2.

Static balance may be obtained by adding a single weight, but there is only one position in which the weight may be put, namely, directly opposite the resultant of the unbalanced static forces, without introducing a new centrifugal couple. This couple when combined with those that may already be present may give a resultant centrifugal couple either greater or smaller depending on the conditions. In the floating-bearing

type of machine the body is rotated and the high spot marked. The mark does not, however, indicate the plane of the disturbing couple; neither is its plane just 90 degrees ahead of or behind the mark obtained. The angle between the two is really a very elusive thing depending among other things on air resistance and the angular velocity of rotation of the body, and had, therefore, best be avoided altogether.

UNITS FOR STATIC AND DYNAMIC UNBALANCE.

Since static balance is obtained by adding a weight whose moment about the axis is equal to that of the disturbing element, it must be evident that the most satisfactory method of expressing the amount of static unbalance is in ounce-inches, that is, as the product of the weight necessary to restore balance multiplied by the radial distance at which it is applied.

Dynamic unbalance is caused by two centrifugal forces each equal, since $a_1 r_1 = a_2 r_2$, to $\omega^2 a_1 r_1/g$, where ω is the angular velocity of the rotating body and g is the acceleration due to gravity. If, as shown in Fig. 2, these two forces act on opposite sides of the axis at an axial distance l apart, they produce a centrifugal couple equal to $(\omega^2 a_1 r_1/g)\ l$, which may be written as $K\ a_1 r_1\ l$. That is dynamic unbalance is equal to some factor K times weight times radial distance times axial distance. The factor K is a variable depending on the angular velocity, but its value must be the same for the disturbing and the neutralizing couple because both necessarily rotate at the same speed; therefore, for convenience, and because it would be difficult to determine its actual value, we may agree to disregard K, whence the dynamic unbalance is proportional to $r_1 a_1 l$, or its unit of measurement becomes inch-ounce inches. Each of the terms of this expression is easily measured, and the result is, therefore, free from all uncertainty that would be involved by retaining the term K.

REQUIREMENTS FOR A SATISFACTORY BALANCING MACHINE.

Until recently there was no machine with which static and

dynamic unbalance could be differentiated. It was assumed that static balance could be satisfactorily obtained by using horizontal balancing ways or their equivalent. Now the quality of the dynamic balance will be good or bad according as the static balancing has been well or poorly done; it is therefore absolutely necessary that the static balance should be perfect before attempting to obtain dynamic balance. Hence the ideal balancing machine should fulfill the following five requirements:

(1) Provide a sensitive and accurate method for determining static balance.

(2) Provide an efficient method for finding dynamic balance.

(3) Indicate the magnitude of the unbalanced force or couple.

(4) Indicate the axial plane of the unbalanced force or couple.

(5) Indicate the sign or direction of the unbalanced force or couple.

So far as the writer is aware, there are only two machines which adequately meet any of these requirements; namely, the Carwen Dynamic Balancing Machine, manufactured by the Carlson Wenstrom Company, and the Combined Static and Dynamic Balancing Machine, built by the Vibration Specialty Company.

CARWEN DYNAMIC BALANCING MACHINE.

The Carwen machine is built in nine sizes having capacities varying from 3/4 pounds to 14,000 pounds and capable of handling objects up to 14 feet long and 60 inches in diameter. One of the intermediate sizes especially adapted for handling automobile crankshafts and small armatures, is shown in the photograph, Fig. 3. Two substantial box pedestals, one at either end of the machine, are rigidly connected by two tie

rods one on either side. The pedestals support a frame similar to a lathe bed. At one end the bed is hinged to the pedestal by a flat plate spring while the other end rests on helical springs. This arrangement allows the free end of the bed to vibrate up and down in a vertical plane and at the same time prevents motion in all other directions. Suspended underneath the vibrating frame is a small electric motor and the two counter balancing weights marked D, D. These weights are splined to the supporting shaft so that a pair of forked arms moved by a right and left screw geared to the hand wheel A, may move them away from or toward each other. When the weights are together they form a symmetrical body in perfect balance; when they are separated a compensating centrifugal couple is introduced.

The body to be balanced is mounted either on adjustable pedestal bearings attached to the bed, or, if it is a chankshaft, preferably in its own base so that it will be supported the same as in actual use. A balanced driving head is fastened to the body, and by means of flexible steel connectors or clips the body and the compensating weights are driven at exactly the same speed and in the same direction by the motor attached to the bed. The shaft on which the weights are placed is driven through a planetary gear to permit the plane of the compensating weights to be brought into the same plane as the disturbing couple in the body. By turning the hand wheel C the angular position of the gear and weights may be altered without stopping the machine.

In using this machine it is absolutely necessary that the object shall be in as nearly perfect static balance as it is possible to obtain by any available method. Neglect of this precaution may lead to erroneous and unsatisfactory results for which the operator has only himself to blame. It must therefore, be assumed that the body is in perfect static balance and that the reading of the machine represents pure dynamic unbalance only.

When the motor is started and the speed is gradually increased, a point will soon be found where the bed has maximum uniform vibrations as shown by the indicator M, which is an ordinary dial indicator graduated in thousandths of an inch. This is the synchronous speed at which the natural period of the springs and the bed is the same as the forced period due to the rotating body. By changing the number of springs, or their characteristics, it is always possible to have synchronism come at a comparatively low speed, say less than 500 revolutions per minute. If the speed is either too high or too low the vibrations will be irregular, periodically increasing and decreasing in amplitude, and perhaps momentarily stopping altogether. By changing the angular position of the compensating weights and their axial distance apart, the position is readily found in which the dynamic unbalance of the body is neutralized.

The value of the disturbing couple in inch-ounce-inches is indicated by a pointer on the dial B. Its plane will coincide with that of the compensating weights, and may therefore be readily found; its sign or direction will be opposite to that of the weights. The machine therefore gives positive and exact information regarding the dynamic unbalance provided the static balance was perfect. It meets all but the first of the five requirements for an ideal balancing machine. It is sensitive and accurate and results are readily obtained, as the adjustments are quickly made without stopping the machine. The compensating weights built integral with the machine restrict its capacity; the machine illustrated has a maximum capacity for neutralizing dynamic unbalance of only 288 inch-ounce-inches. To increase the range by using heavier weights would make it less sensitive.

VIBRATION SPECIALTY COMPANY'S BALANCING MACHINE.

The appearance of the Vibration Specialty Company's Combined Static and Dynamic Balancing Machine is shown in Fig. 4. It will be noted that the unit is relatively small and

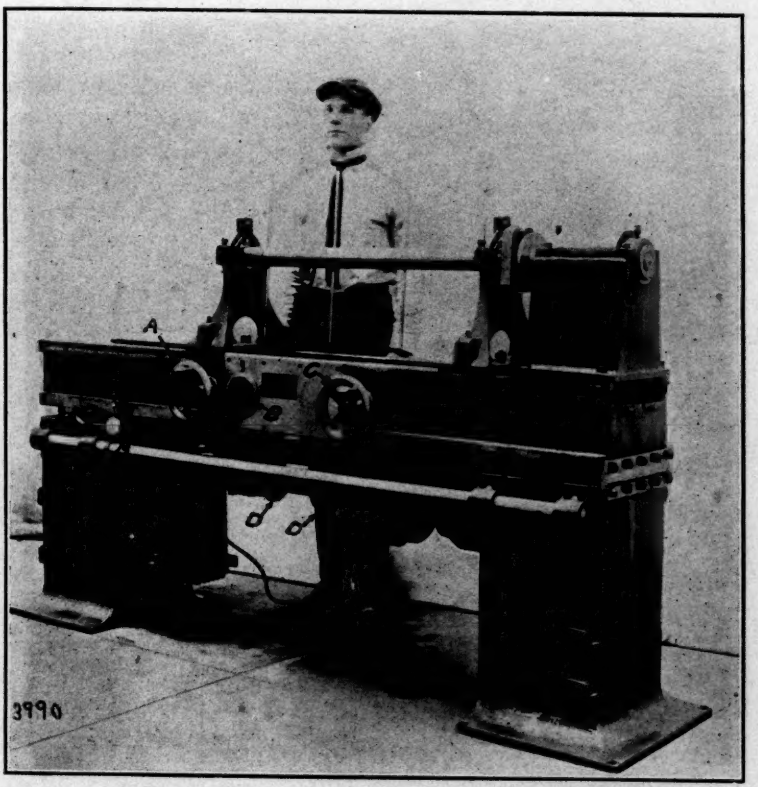


FIG. 3.

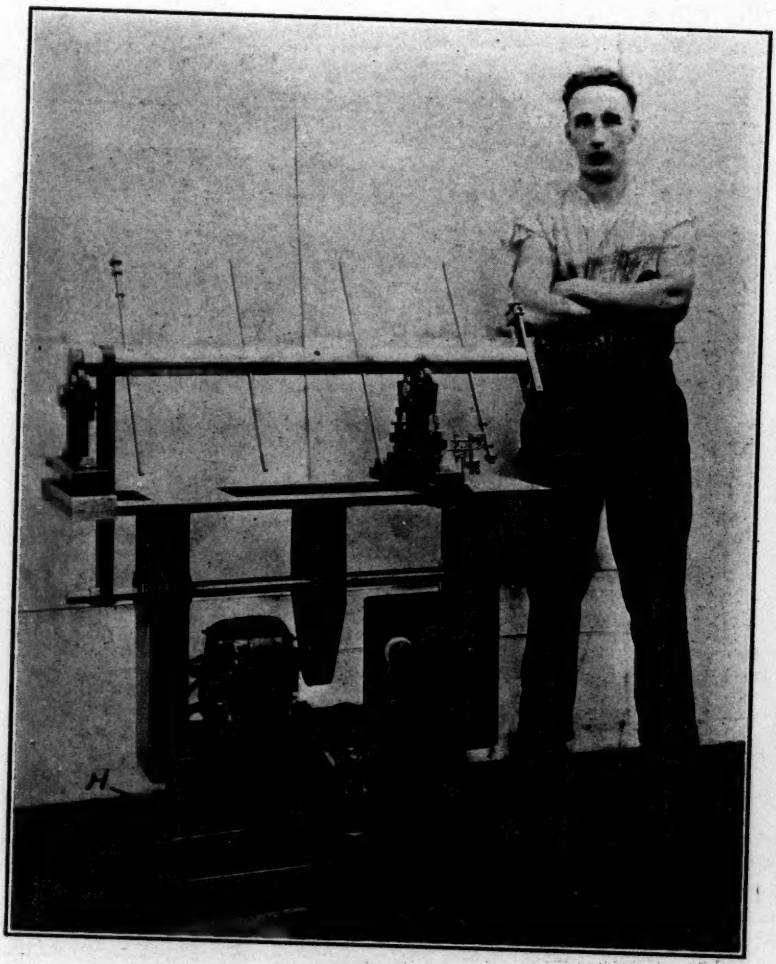


FIG. 4.

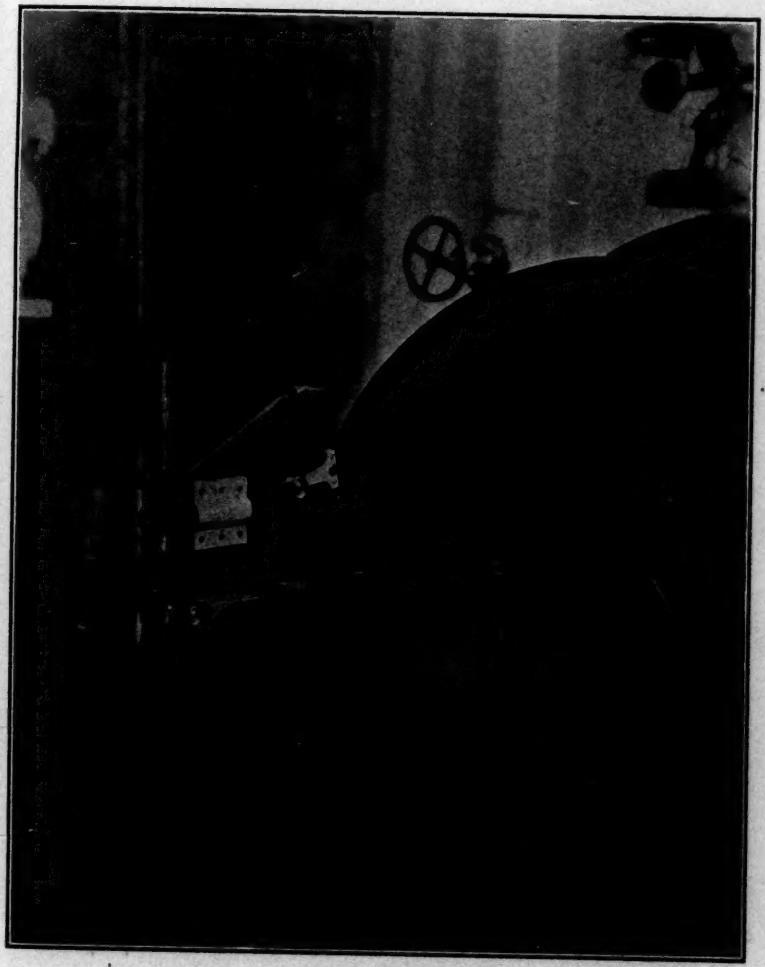


FIG. 5.

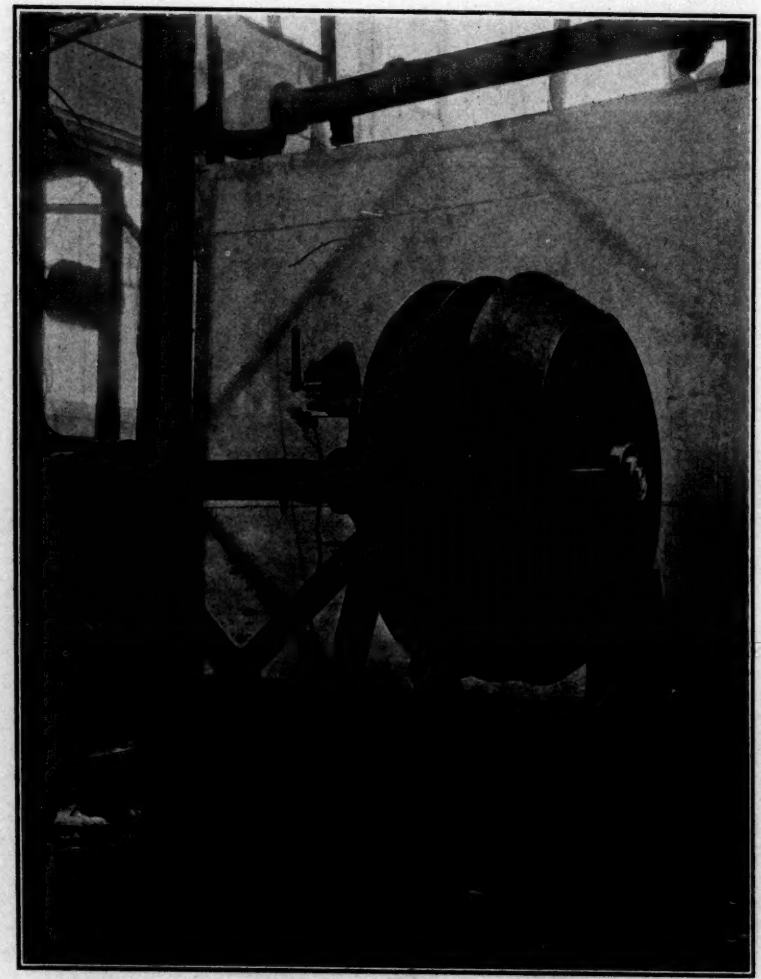


FIG. 6.

compact and simple in construction. A cast-iron box base supports a vertical frame by means of a steel-plate hinge H which permits the frame to vibrate horizontally in one direction only. An arm extends out from each side of the frame and carries at its end an adjusting screw and a hand wheel. By means of the hand wheel the tension of a helical spring beneath each arm may be adjusted in order to increase or decrease the natural period of vibration of the frame. The adjusting screws are also used for rigidly locking the frame when the machine is used for dynamic balancing. The frame carries a small motor with its starting and speed control rheostat. The countershaft just above the motor is driven by a chain drive. A rubber belt at one end of the counter shaft is used to drive the object to be balanced.

The frame carries two pedestal supports on which the object to be balanced is mounted on rollers. The left-hand pedestal is rigid, thus differing from the right-hand one, which is supported on a thin phosphor-bronze plate hinge in the same manner as the frame is attached to the base, and which in the same way permits vibration of the pedestal in one direction only. An arm also extends out on each side of the pedestal, with adjusting screws and small helical springs for changing the natural period of vibration of the pedestal, or for locking it rigidly in position when determining static balance.

Two dial indicators are shown at the right-hand end of the machine. One of the indicators is supported by the frame and shows the amount of vibration of the pedestal; the other is supported from the base of the machine and indicates the vibration of the frame itself.

It will be observed that no compensating couple is used in this machine; instead a balancing clamp is attached to one end of the object to be balanced, and hence, rotates with it. The balancing clamp is shown in position on the test arbor in Fig. 4. The construction of the clamp is very simple; it may readily be fastened in any angular position and its neutralizing effect changed by moving the sliding weight nearer to or far-

ther from the center. As shown the weight is nearly all the way out, where it has its maximum compensating effect, and the clamp is in the proper plane to neutralize the centrifugal couple due to the small weights on the ends of the two arms of the test arbor. In using this machine the first step is to check the static balance of the body under test. For this purpose the pedestal support on the frame is rigidly locked and the frame is released from the base so that it may vibrate freely due to any unbalanced static force which will tend to cause a translation of the body, as in Fig. 1. The motor is then started and the body rotated, the speed is varied, and, if necessary, the tension of the springs changed, until the vibration as shown by the indicator becomes a maximum. The clamp is then applied and its position and that of the sliding weight shifted until the resultant of the unbalanced static force is neutralized and the vibration reduced to zero. The position of the clamp then shows the plane of the resultant of the unbalanced static forces while its value in ounce-inches is given by the weight in ounces of the sliding weight multiplied by the distance it has been moved from its neutral position on the clamp. The correction is made by adding or removing weight at one point; preferably, in the transverse plane through the center of gravity of the body.

After the body is perfectly balanced statically, the frame is rigidly locked to the base by the arms and adjusting screws, and the right-hand pedestal is unclamped. The body is again rotated at various speeds until the synchronous speed, at which the vibrations are a maximum and uniform, is found. The balancing clamp is then applied and the positions of the clamp and of the movable weight are shifted until the dynamic unbalance is neutralized and the vibration becomes zero. The unbalanced centrifugal couple is then numerically equal to the weight of the movable weight multiplied by the radial distance it was moved from its neutral position multiplied by the axial distance from the central plane of the clamp to the center of the rollers on the left-hand rigid pedestal. The position of

the clamp shows the plane and the sign of the disturbing couple, which may then be neutralized by adding or removing weight at two points in the same axial plane on opposite sides of the axis.

The Vibration Specialty Company's Machine meets all five of the requirements and does so in a really simple manner, much less complicated than might be supposed from the somewhat involved description just given.

It is, of course, rather inconvenient to stop the machine each time the position of the clamp or the neutralizing weight is to be changed, and also to obtain the amount of the unbalance by measurement and computation. An experienced operator, however, usually does not have much difficulty in quickly obtaining a satisfactory result. On the other hand, the balancing clamp possesses certain valuable features which in the writer's opinion more than outweigh any disadvantages it may have. If the body is badly unbalanced so that a single clamp is insufficient, two or more may be used; moreover, these need not be of the same size, thus one may have a small movable weight, or even two weights, one large and the other small, so that the final adjustment may be accurately made with the small weight.

The machine is exceptionally sensitive and accurate for both static and dynamic balancing. For static balancing in particular the machine is far more sensitive than balancing ways; this is of great importance because good dynamic balance cannot be obtained unless the static balance is first good. Dynamic unbalance has no effect on static balance, but the reverse is not true.

When arranged as in Fig. 4 the machine is adapted to handle objects up to 20 inches in diameter, but by the addition of certain auxiliary attachments it is possible to use the same machine for objects up to 60 inches in diameter and weighing, say, up to 1,000 pounds. With such large objects it is necessary to use a large balancing clamp. Hence, by using the auxiliary attachments and providing a number of balancing

clamps of different sizes the machine may be adapted to handle a large variety of work. Turbine rotors weighing up to sixteen tons have been successfully balanced on larger machines built by the Vibration Specialty Company.

LIMITS OF BALANCE.

With satisfactory balancing machines available, with one of which at least it is possible to distinguish between static and dynamic unbalance, it seems desirable that certain limits or tolerances for unbalance should be established. These limits should be included in the specifications for the various types of machinery. Just what they should be probably no one now knows, and they can be fixed only by experience and a thorough investigation of the entire subject. Undoubtedly, the limits will vary, not only for various classes of machinery, but also with the size and weight. Mr. Akimoff has proposed tentatively that for bodies weighing not over 150 pounds and with journal diameters of not over three inches the limit for static balance should be 0.4 ounce-inches, with a corresponding limit for dynamic balance fixed at 12 inch-ounce-inches.

AMPLITUDE OF VIBRATION IN UNBALANCED BODIES.

Obviously if a rotor is perfectly balanced it will set up no vibrations. If the balance is not perfect there will be more or less vibration, and sometimes it is difficult to determine whether or not the amount of vibration is excessive. Often an inspector is in doubt whether to pass or reject a machine because it vibrates at certain speeds. He knows that the manufacturer balanced it as well as he could without using a balancing machine constructed on a sound theoretical basis, and besides he has no way to determine the actual amplitude of the vibration, he can only judge that it is good or bad by the "feel." Or perhaps, it is necessary to improve the balance of a unit already installed; weights are added here and

there, the unit is brought up to speed, and quite likely a discussion ensues between different observers as to whether the balance is better or worse than before.

To answer these perplexing questions the Vibration Specialty Company has recently developed an instrument called a Vibroscope. The first one built was recently tested at the Engineering Experiment Station with interesting results. The Vibroscope is shown in Fig. 5, attached to the outboard bearing of a horizontal turbo generator. It is built on the principle of the seismograph; the frame or vibrating part of the instrument is arranged so that it may be conveniently attached to the machine whose amplitude of vibration is to be determined. The upper leg of the frame is slightly inclined and has attached to it by a thin, flexible spring hinge a heavy, flat-plate pendulum supporting a micrometer microscope at its outer end. This pendulum plate has a very slow period and becomes the "steady point" of the arrangement. Directly beneath the microscope at the end of the horizontal part of the frame is a narrow slit illuminated by an electric light. By looking through the micrometer microscope with the instrument at rest the width of the slot is adjusted to, say, five divisions on the illuminated scale; then with the machine running any apparent increase in width of the light slit is due to vibration. By calibrating the instrument the relation between the reading of the Vibroscope and the amplitude of the vibration may be determined.

To obtain some idea of the vibration of actual machines the Vibroscope was attached to two turbo-generators as follows:

(a) 300-k.w. turbo-generator set. Speed 1,800 r.p.m. This unit operates satisfactorily and with very little vibration. The outboard bearing at the generator end vibrated about 0.0001 inch. The turbine shell in a horizontal diametral plane vibrated 0.0003 inch.

(b) 75-k.w. turbo-generator set. Speed 2,400 r.p.m. This unit runs badly, commutation is poor and vibration excessive, particularly, at the main bearing between generator

and turbine. When attached to the end bearing as shown in Fig. 5, the Vibroscope indicated a vibration of 0.0003 inch. The main bearing, however, had an amplitude of 0.0026 inch. It appears, therefore, that for turbo-generators a vibration of 0.0005 inch is permissible, but that 0.0025 inch is not. Where the limit lies between these two values can be determined only by further tests.

BALANCING A DESTROYER FAN.

A 30-inch destroyer fan was selected for test to determine the amount of static and dynamic unbalance and the decrease in the amplitude of vibration due to the elimination of each. The fan, shown in Fig. 6, had an overall diameter of 44 inches, and weighed 555 pounds. The manufacturer had evidently balanced the fan statically as well as his facilities permitted, as was evidenced by a weight at the periphery of the wheel in the plane through the center of gravity.

The balancing was done on the Vibration Specialty Company's machine shown in Fig. 4, with the auxiliary attachments previously referred to. The static unbalance was found to be 48.4 ounce-inches, which is equivalent to 2.2 ounces at the periphery of the fan; the dynamic unbalance was 1,400 inch-ounce-inches, equivalent to 6.2 ounces at the periphery and in two transverse planes 10.6 inches apart axially.

The balanced fan was then mounted as in Fig. 6, and driven by an electric dynamometer through a flexible coupling. The Vibroscope was attached to the pedestal bearing adjacent to the fan. At 1,425 revolutions per minute, the operating speed of the fan, the amplitude of vibration with the fan balanced was 0.0035 inch. By removing the fan from the shaft it was found that the vibration of the shaft alone was still 0.0035 inch at 1,425 revolutions, due to unbalance in the dynamometer and especially to a heavy unbalanced shaft coupling adjacent to and behind the fan. Hence it is reasonable to conclude that the vibration observed with the balanced fan was not due to

the fan itself. When the two dynamic correction weights were removed, leaving the fan in perfect static balance, the amplitude of the vibration increased to 0.0068 inch. Finally after removing the static weight also, thus restoring the fan to the condition in which it was received from the manufacturer, the vibration became 0.01 inch.

The conclusion is obvious. In this case, the manufacturer had balanced the fan, and, as such things go, it did not seem so very bad. But after securing static and dynamic balance the vibration dropped to one-third of its original value, and would undoubtedly have been even less if the supporting shaft had been in balance. Such an improvement is surely well worth while.

In closing, just a word of warning. Although the importance of securing static and dynamic balance can scarcely be overestimated, it must not be thought that vibration is always due to a lack of one or the other. Other disturbing factors, such as torsional vibrations, may be present and must often be taken into account if the problem is to be fully solved.

BALTIMORE AND SOME EARLY YEARS IN THE ENGINEER CORPS, U. S. N.

By Chief Engineer Fred. G. McKean, Member.

The following notes, in their departure from the usual articles in the Journal of the American Society of Naval Engineers as to solid science and the description of recent attainments, call for some explanation.

It is believed that the History of the Corps should be written before the steam engine is possibly superseded by electricity, aviation, or whatever is in the womb of the future; and that when it is written, all details which will throw light upon its affairs will be useful, varying as they might, from Passed Assistant Engineer Bennett's monumental work, the "Steam Navy of the United States," through various other writings to the personal reminiscences of men some of whom may be about to join the majority.

Two recent articles in the Journal, emphasize the advisability of such action. The instructive review "Fifty-Year Retrospect of Naval Marine Engineering," by Rear Admiral C. W. Dyson, U. S. N., showed even those who have some memory, that it is refreshing to take stock once in a while of events which have passed; and the interesting summing up in another article sub-entitled, "A Quarter of a Century of the Journal," by Rear Admiral John R. Edwards, showed the danger of delay, for in the preparation of that article it was found that several important points of fact had already become matters of doubt, and necessitated reference to private diaries and to comparison of sometimes treacherous recollection.

If the foregoing remarks are conceded, it remains to state more specifically that the intention of the present review is to mention briefly a few occurrences and persons known in

"the fifties" of the nineteenth century, but also digressing occasionally to times before, and alluding now and then to the almost incredible advances made since then.

Baltimore for several reasons played an important part in the progress of the Engineer Corps, and, in fact, of the United States. Its proximity to the Capital, the production of Government work by its five best known machine shops, its situation as the headquarters of the whole South and connecting link with the whole North, combined to give it prominence; while historically, the previous aid of western Maryland in casting cannon, the operation of well-known furnaces, a rolling mill which later furnished the armor plates of the first Monitor, Rumsey's experiments in early steam navigation on the upper Potomac, the fame of Baltimore Clippers in peace and war, and its achievements almost as a pioneer in railroading and telegraphing, gave it a traditional and deserved pride.

For the purposes of the present compilation it may be stated that among the machine shops of Baltimore the Vulcan Works, at that time under the management of James Murray and Henry R. Hazlehurst, successors of Watchman and Bratt, not only did much Government work—marine, lighthouse, bridge, railing, etc., but included among its other opportunities candidates for entrance into the Engineer Corps, insomuch that the modest and rather dingy drafting room became the rendezvous of several successive groups of aspirants and grades of naval engineers and some outsiders, who brought new questions and contributed matter for inquiry and general profit. The chief draftsman, Mr. Eliakim T. Robb, was unusually fitted to help candidates, being thoroughly competent, able and always cheerfully willing to impart information. Furthermore, the questions asked and answers given at the examinations were being eagerly collected by industrious candidates, and later became bases for monographs upon the various branches, and in at least one instance used after the war as a lecture in the Engineer Department at Annapolis.

Baltimore, in common with the engineering world, discussed warmly and intelligently the problems of the day; thus, it may surprise some readers that the question of the possible rejection of surface condensation in seagoing steamers, the disbelief of many in the Giffard injector, the pitting of boilers, the free access of water to propellers, the use of superheated steam, rotary engines, high pressure and other matters now held as settled, were by no means so held at the time of which we write.

For the information of very young engineers and the refreshing of some older ones, it may be recalled that doubts of surface condensation based their uncertainty upon the "crawling of materials," whereby condenser tubes moved until the ends were exposed, and not only so, but upon the "cussedness" of the material in not compensating the action by resuming the original position when cold.

As to the Giffard injector, now almost indispensable in locomotives and very useful for stationary boilers, the idea that a pipe leading from the top of a boiler to the water space of the same boiler, could feed water into it, seemed so repugnant to some Baltimore men (and they were supported by the bulk of European engineers) that they declared it impossible; and it should be recollected that the inventor, being unable to explain the action, contented himself by inviting unbelievers to come and see it in operation: then came the change—science having seen the fact, remembered that it was entirely according to physical law.

There are at least two mistaken ideas about the Engineer Corps of the Navy at the time herein referred to: one is that there was almost no information in print; the other that the older Chiefs, outside their profession, were almost ignorant men. As to the former, the status of marine engineering and the requirements of the service did not call for very much, but what was available was generally sufficient. We have mentioned the collection in manuscript of examination questions and answers, and certain books of the day were well

known and eagerly used; among which may be mentioned Byrne, followed by Bourne on the Steam Engine, Main and Brown on the Marine Steam Engine, Tomlinson's Mechanics, Gregory's Mathematics for Practical Men, Bonnycastle's (much-thumbed) Mensuration, Legendre's and other Geometries, Bourdon's and other Algebras, Eubank's Hydraulics, "as good as a novel," Lardner's somewhat discredited Natural Philosophy, Overman on the Manufacture of Iron, Johnson's Report on Coals, Nystrom's (the predecessor of Haswell's) Engineers' and Mechanics' Pocket Book, and several more; while in Baltimore at least, the libraries of some of the iron works, and more especially that of the Maryland Institute, contained books of great utility; the lectures of the latter on Chemistry and Philosophy were instructive, and its evening classes in drawing, etc., were open; engineers of the navy and of steam boats were always willing to impart the fruits of their experience; there were few mechanical periodicals, but those which existed brought new inventions and discoveries to notice and presented problems for study. Of course there was nothing like the mine of publications with which we are familiar now, with essays on every detail of every branch of marine engineering, and certainly such articles as the latest specifications in naval machinery and the papers found in the files of our JOURNAL, would have been invaluable to students in those early days, containing, as they do, the results of long labor in practice and theory.

The other statement, that the old chiefs could shove a file and chip to a line but knew little else, was also largely an error. Mr. Isherwood was well known to the whole scientific world, and others of his contemporaries were familiar with one or two languages or had dipped into the higher mathematics.

The Vulcan Works of Baltimore had done much naval work for the Government, including engines for the second *Princeton* and the *Susquehanna;* the firm competed during the time we are treating of, for the machinery of the *Dacotah,* one

of six vessels intended to be fast: the *Iroquois* was probably the best, but the Baltimore ship was not far behind her in speed and efficiency; however, the great weight of her geared engines and the care of Mr. Murray in making the details heavy enough to be safe, and perhaps the conscientiousness of the firm, contributed to the failure of the company in business. The *Dacotah* is worthy of notice for several reasons; among them the split connecting rod and main shaft revolving within it was a peculiarity, also the condenser had rather short perpendicular tubes, the upper ends expanded and the lower ones all packed with one follower-plate; some of the condenser features are almost obsolete now, but it was so successful in returning the water of condensation over and over again to the boilers that the latter developed one of the first mysterious cases of "pitting," an evil so little known at that time that it was universally attributed to the supposed presence of copper in the rain-water tanks at the Norfolk Navy Yard, where the vessel was fitted out. It was never dreamed that for years the scientific world, backed by committees appointed by the Lords Commissioners of the Admiralty, would later experiment upon and wrestle with the question of similar deterioration in boilers, and almost despair of its solution, in spite of at least three voluminous government reports in Great Britain, with colored plates, photographs and a wilderness of testimony, expert and otherwise.

Marine practise of the time referred to in our introductory remarks would have few advocates now, and deservedly so; the beam-engine was still holding its own, even for seagoing purposes; in boilers the advantages and disadvantages of copper and iron as materials, and of vertical and horizontal tubes, differential tubes, and even flues, were still discussed; Hall's, Sewell's, Stimers' and Pirsson's (double-vacuum) condensers were used, with others that are mere names; Stevens' and Sickel's cut-offs for poppet valves were best known; the Griffith, Hirsch, Mangin and other propellers were recommended, and some grades of naval engineers had to sketch and describe many devices which are forgotten now.

Drafting, previous to "the fifties" was ordinarily coarse in line, because among instrument makers even Hommel-Esser of Aargan in Switzerland, was only building up a reputation and his best productions (brass-mounted) were not pleasant to handle; but there were Department draftsmen in Washington, notably under the Lighthouse Board, who turned out beautiful work, which aroused admiration and emulation in the Baltimore contingent of the Engineer Corps, soon developing several young men skilled in the art. But the topic of drafting, in and out of the Navy, would itself need an article, and would testify to the excellence, among others, of George B. Whiting and John Ericsson, and their important relations in after years with the Bureau of Steam Engineering.

An account of Baltimore in some early years of the Engineer Corps, would not be complete without mention of the personnel who met in the drafting room of the Vulcan Works, many on the threshold of their career, but some even then far advanced in years and rank; two of the latter being known in warm admiration of their kind helpfulness as "Daddy" Williamson and "Pappy" Stewart, of Virginia and Pennsylvania, respectively. There was a spirit of comradeship among the aspirants, students and other hard workers for the few vacancies to be filled, but every one helping the whole by handing in his store of findings to the common stock. Mr. Bennett's list includes the whole Corps down to 1893, and he enlarges upon the Engineers-in-Chief and others of note, but without special points in reference to some of the juniors, and it is with apologies for necessary incompleteness, unintentional forgetfulness, etc., that we approach this part of the subject. Shock was already a well-known Baltimore name. Among the early Assistants best known, but even then by legend and anecdote, was J. Bolivar Knox, always called Bolivar, a name very fashionable at that time in honor of "The Liberator."

Ed. Chassaing was one of the brightest little men the Corps ever possessed, becoming fleet engineer in the civil war at an early age, and among other things pointing out an error in

King's "Steam," etc., upon the true reason why the salinometer stem was unequally divided; he resigned and later died of yellow fever in South America. Bob Schley and George Riley were authorities upon nautical affairs generally, having made voyages in sailing vessels; the former died in the service, the latter was afterwards connected with the Copper Works on the Gunpowder River in Maryland. Wm. Thompson and a few others "seceded," and advanced in the Confederate engineer corps. Wilkins Cragg did not remain long in the service after the war, but Louis J. Allen did. Jim Noble was a sturdy, conscientious hard worker. Tom Bordley was a good draftsman and a good man. Tom Dukehart resigned. Phil. Voorhees was an accomplished engineer and gentleman, and later went into the patent law profession. John Wilson, genial and deservedly popular, resigned after the civil war. Al. Murray, a bright member of the Corps, was blown up in the *Chenango* boiler explosion, upon her trial. Nick Littig, light-hearted, seemed to make himself appear frivolous, but when, after the war, the *Oneida* was sunk in a collision, Chief Engineer Littig stuck to his post below and went down with the ship—a death as noble as man could wish.

Having mentioned briefly a few of the once well-known names of Baltimore naval engineers of the time specified, the writer feels it incumbent upon him to refer to a small number of a rather numerous class of civilian engineers, chiefs and assistants of seagoing and river steamers, superintendents and workmen in machine shops, and others from whose ranks the engineer corps obtained a goodly number of volunteers or acting appointments during the civil war, and who did excellent service in spite of the prejudices and teachings of their surroundings ashore and afloat: their history has perhaps never been, and probably never will be written, deserved though it is. Among a few names, but not all eligible for the Navy, may be mentioned Andy Miskimon, second to none when it was a question of running the lines in a hull and erecting engines therein; and here it may be interjected that there

was nothing in English print on the subject at that time. Other engineers were Tom Roberts, a willing helper; the older Gragg, a well-read man; Platt; Tim. O'Connor, who had run the first steam fire-engine in Baltimore; Rube McClanahan; the Lawrence and Thornton brothers; Bill Tipton, a finisher and afterwards an Acting Engineer in the Navy, who rallied a party of workmen from the Vulcan and Reeder Works, scaled Federal Hill just behind those shops, captured a cannon with which some fellows of the baser sort were firing a salute in honor of a Confederate performance, dumped the gun into the harbor, and showed that some Baltimore mechanics, at least, had not bowed the knee to the Baal of disunion; there were also John Cahill, John Menshaw, Bill Tolson and Tom Smith, the last the boss boiler maker.

Some events of the great war of 1914–18 may be recalled briefly and with a purpose, though not always having a bearing on naval engineering. The British have mentioned with deserved pride their saving of time by running troop trains directly on board special vessels which carried them across the Channel; and yet some Baltimore engineers will remember that before the fine bridges were built across the Susquehanna, the passenger trains to Wilmington were carried across the river upon large ferry boats. The use of parachutes has been spoken of as almost the last word in aviation, and yet, at the time covered by the title of this article some youths of Baltimore were thrilling over the fascinating book of John Wise, the Pennsylvania aeronaut, who, first in an emergency to avoid his balloon being carried out to sea, and afterwards by daring choice, turned his balloon itself into a parachute, by throwing out ballast until he was carried into such a rarified atmosphere that the balloon burst, and after a terrific fall of some distance, the bag, caught by the cords, became to some extent an umbrella and landed him safely; the flight across the Atlantic, now imminent and certain, was even then a theory.

We have read with pathetic interest the doings of faithful dogs and pigeons which carried war messages, and yet Scharf, in his Chronicles of Baltimore, relates how, first between that city and Washington, and later to Philadelphia and New York, a "pigeon express" was established before the telegraph superseded it.

Nowadays superheated steam is almost a commonplace necessity, and yet when the civil war broke out, an assistant engineer of the Navy was taking data and indicator cards by permission, on a boat running out of Baltimore, in order that the true economy in superheating should be tested. We boast now, and very justly, of carrying higher steam than our predecessors, and 400 pounds seem to be coming: at the time included in these notes, copper boilers still had their advocates, with ruder methods in riveting; but even then the book, Alban on the High-pressure Engine, was studied with an eye to the future, and there were 140-pounders on the Mississippi River. It is true that one of these would explode occasionally when an ambitious captain raced for the privilege of carrying on his steamboat the horns which denoted championship, but most human attainments must pay their price in human life. Torpedoes of a kind were known in our civil war, and registered a hit now and then. Oxy-acetylene welding almost worked miracles in repairing the machinery of interned men-of-war thought to have been irretrievably destroyed during the recent conflict, and yet it is many years since a United States naval engineer (from Baltimore, by the way) insisted that the pieces of a broken casting should be "burned" together instead of a new casting being made, though a well-known machine shop in the Far East declared that it had never heard of the process and would not be responsible for failure.

The turbine has almost revolutionized marine engineering, and yet rotary pumps were known in "the fifties" of the nineteenth century with yearnings toward less friction and waste of steam, while the pumps at least marked an improvement on the reaction globe of Hero of Alexandria. It must be con-

ceded, however, that the compound engine about to be revived was somewhat of a theory then, while multiple compounding, the electric drive, internal combustion, etc., were hardly in the mind of man. The comparative statements above are not intended to detract from modern achievements: the writer yields to none in profound admiration of progress in what is only the beginning of the twentieth century, but they are reminders of the old saw that "History repeats itself in the most unexpected circumstances," and they leave one in amazement of what the future may yet have in store for us.

The work of the writer is nearly finished: it is imperfect, rambling and inadequate, but if it will induce more competent hands to record the relations of various places to the early or late history of the Engineer Corps it will have accomplished good. And there is great store of interesting facts, from Baltimore, in extension of the foregoing lines; from Philadelphia, called slow but not so in engineering matters; from New York, based possibly on the local history by ex-Engineer-in-Chief Haswell, "the first engineer in the United States Navy," but to be rewritten with a slant toward the Corps; from Boston, for New England furnished us with some of our most intellectual members; from Washington, with wave upon wave of selected designers, draftsmen, calculators, workers and administrators; and from other localities, and the facts should be chronicled before it is too late.

But whatever may come, we are all confidently hopeful, as we have every reason to be, that the Engineer Corps of the U. S. Navy will live up to its traditions, and will continue to possess a Journal worthy to register its progress. May Corps and Journal be perpetual!

NOTES ON STEAM TRAPS.

By F. G. Hechler, Associate.

The requirements for general marine service are unusually severe; the trap must function satisfactorily at any pressure from atmospheric to the highest boiler pressure carried; its operation must not be adversely affected by the motion of the ship in a seaway; the *valve mechanism* must be simple and readily accessible for inspection and repair without having to disconnect any pipe line; the *discharge valve* must not be at the bottom of the trap where scale and sediment collect.

The body of the trap should be heavy and rigid, and flat surfaces should be avoided whenever possible. If the trap is intended for use with high steam pressures, the requirements of the American Standard for extra heavy flanged fittings should be complied with in determining the thickness of the body and of the cover flanges, as well as the size and the spacing of the cover bolts. Comparatively wide gasket spaces should be used to prevent the gasket from blowing out. Counter-bored recess gasket flanges are usually most satisfactory. The inlet and outlet opening should be on the body of the trap so the cover may be removed without breaking any pipe lines, and there should be a bottom blow-off for cleaning the trap of scale and sediment.

In ball float traps the buoyancy of the float must be sufficient to open the valve against the steam pressure. In this case the following relation must be satisfied:

$$W + (p_s - P_d)\ a < F,$$

Where W = weight of float in pounds p_s = steam pressure in trap, pounds per square inch gage; p_d = pressure against which trap discharges (usually atmospheric), pounds per square inch gage; a = contact area of discharge valve, square

inch, F = effective buoyancy of float = displacement in cubic feet times density of water in trap at temperature corresponding to the pressure, pounds per cubic feet.

A 1¼-inch trap of the simple float type and intended for use with pressures up to 300 pounds, had a float displacement of 0.1278 cubic foot; the float weighed 4.17 pounds. The maximum size of the discharge valve calculated from the relation just given is $4.17 + (300 - 0)\, a < 0.1278$ times 52.6, or $a < 0.0085$ square inch, and the corresponding valve diameter is 0.104 inch.

Small discharge valves are unsatisfactory because they are liable to become clogged with scale and because their discharge capacity is too small. Large floats are not desirable because they increase the size of the trap unduly, and because it is difficult to make a large float to withstand the high pressures. For these reasons the use of levers to increase the float pull is usually resorted to.

The effective pull of a bucket float for opening the valve may be found by filling the trap and bucket with water, attaching a spring balance to the valve stem and measuring the pull necessary to lift the bucket. Tests have shown that for bucket floats the relation between maximum operating pressure, valve area and bucket pull may be written as $p_a \times a^n = F$, where p_a, a and F have the meanings already given, and where the exponent n has a value that is usually slightly less than unity (results varying from 0.93 to 1.00 have been found). When graphed on logarithmic paper this curve is a straight line whose slope gives the value of the exponent n. The relations between p_a and a as determined experimentally for a line of bucket-float traps varying in nominal size from ¾-inch to 2-inch are shown in Fig. 1. The p intercept for an area of 1 square inch is the pulling power of the bucket as determined with the spring balance. In practice the valve areas should be somewhat less to insure satisfactory operation for slight over-pressures that may occur, and also to per-

mit grinding in the valve and trueing up the seat when necessary. The allowance in area should be from 10 to 15 per cent; valve areas determined in this manner will be consistent

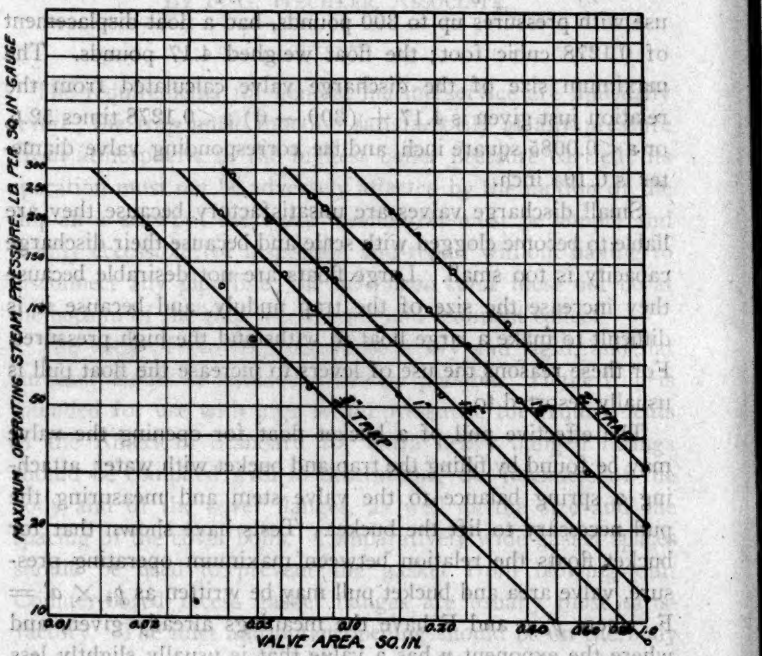


FIG. 1.

and will give the maximum safe capacity. The design of the bucket float requires care; the addition of a small weight at the outer end of the bucket may increase the maximum operating pressure of the trap with a given valve as much as 20 per cent.

TRAP RATINGS AND CAPACITIES.

There is no logical reason why steam traps should be rated, and also usually purchased, on a basis of the size of the inlet and outlet connections. Traps of the same nominal size may

have widely different capacities at the same pressures as shown in Table 1. When specifying capacity it is necessary to state the pressure at which it is required. Capacities should always be determined with hot water delivered to the trap by steam at the necessary pressure; capacities determined with cold water and a pump will be too high.

Steam traps (except expansion) will operate satisfactorily at any pressure lower than the designed pressure without change or adjustment, but with a smaller capacity. To secure maximum capacity the valve should always be adapted to the operating pressure. When a trap is likely to be used over a considerable range of pressures it should be provided with several interchangeable valves of different size. For pressures up to 300 pounds it has been found sufficient to use three valves and seats designed for pressures of 30, 150 and 300 pounds. Table 2 gives the capacities as determined by test of a line of bucket traps at pressures of 30, 100 and 200 pounds. These capacities are the maximum quantity the traps can pass when discharging continuously under full steam pressure.

INSTALLATION, CARE AND OPERATION.

When installed on shipboard, traps are seldom horizontal, and the constant change in inclination, due to the pitching and rolling of the vessel in a seaway, may interfere with their action. Many of the bucket and ball-float traps lose their water seal and open the discharge valve, permitting steam to blow through when inclined in a longitudinal direction through an angle of about 15 degrees with the horizontal. This difficulty is to some extent obviated by making the operating mechanism as compact as possible; it has also been found with bucket traps that it is possible to increase the safe angle of inclination up to about 25 degrees by making the bucket with an inclined upper edge similar to that shown in Fig. 4, page 244, JOURNAL A. S. N. E., May, 1918. Whenever possible,

traps should be installed on shipboard with their longitudinal axis fore and aft, thus decreasing the liability of steam blowing through. The operation of the traps shown in Figs. 6 and 7, page 243, JOURNAL, A. S. N. E., May, 1918, is not affected by any inclination likely to occur. Expansion traps are also free from this defect as they operate equally well when installed in any position.

When the pressures are subject to considerable variation, expansion traps are not recommended for use unless provided with an automatic pressure-compensating device. With some traps of this type it is difficult to maintain a given setting even under constant pressures, and in some cases the temperature of the surrounding air has an appreciable effect on the operation. Expansion traps may be adjusted to give either a constant or an intermittent discharge; they do not become air-bound, but sediment and grease prove troublesome. Since they have no storage capacity, 8 or 10 feet of pipe should be provided for cooling the condensate somewhat before it reaches the trap.

When continuous-discharge traps are operating at low capacities the *wire drawing* at the valve is likely to prove destructive to the valves and seats; these should be arranged so they may be readily examined and renewed when necessary. The continuous-discharge trap is well suited for use with long discharge lines, preventing water hammer and sudden shock likely to occur with the intermittent type under this condition. Whenever possible traps should be located so that the condensate will flow to them by gravity. If such a location would make the trap inaccessible, it may be placed at a higher level if the line to be drained carries a pressure of 5 pounds or more. The drain line should always be connected to the lowest part of the line or apparatus to be dripped. A trap should be connected to drain only a single line unless the pressures are known to be identical; this is a rare condition and is likely to be disturbed from time to time. Traps may

discharge to a higher level than the trap if necessary; it is usual to assume that the condensate may be elevated two feet for each pound of excess pressure in the trap over that in the discharge line. Traps should always be connected up with unions and fitted with a by-pass so that the trap may be overhauled without interfering with the proper drainage of the line. A test valve or cock should be placed in the discharge line close to the trap to determine how the trap is functioning. A blow-off line not smaller than ½ inch should be provided for cleaning the trap.

A regular *inspection* of all traps should be insisted upon, and they should be promptly overhauled when improper functioning is detected. Steam traps are prone to leak due to cutting of the seat or scale lodging under the valve. Because under these circumstances there is no difficulty due to an accumulation of water, repairs are often neglected, permitting a serious steam leakage to occur.

FITTINGS.

Gage glasses are desirable aids for determining the condition of a steam trap. Under high steam pressures tubular gage glasses sometimes break frequently so that they are often dispensed with entirely. Gage glass fittings should be of the ball-check type so they will be self-closing in case of accident. An air cock is necessary with most of the float traps to relieve the air when starting, and at such other times as may be necessary. A removable strainer placed at the trap inlet, or in the pipe line just before the trap, is often of value in preventing scale and sediment reaching the working parts of the trap.

TESTING.

No uniform code for *testing* steam traps has been adopted. New types of steam traps submitted for use in the Naval Service are subjected to the following tests (see JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. 28, No. 1,

February, 1916): (a) *Hydrostatic pressure* test of 150 per cent of the maximum working pressure (ball floats are separately tested to 500 pounds per square inch. (b) *Functioning* test with steam pressures from 25 to 300 pounds gage. (c) *Capacity* test to determine the maximum quantity of water the trap can discharge under various steam pressures. (d) *Rolling* test with the trap mounted on a rocking platform to simulate the motion of a vessel in a heavy seaway. (e) *Endurance* test with the trap installed for at least three months in the power plant to determine whether it will function properly for a reasonable length of time.

The specifications of the Bureau of Steam Engineering, Navy Department, for steam traps are as follows: (1) Traps will be of either open-bucket or ball-float positive action type, unless otherwise specially approved by the Bureau, so designed that they can be overhauled without disconnecting the pipes, and the bonnets removed without disturbing the lagging. They will be located where they are easily accessible for examination; (2) the trap must be of a type that has given satisfaction in actual service on board a Naval vessel, or passed a satisfactory test at the U. S. Naval Engineering Experiment Station, Annapolis, Md. If a ball float is used the ball must be able to stand an external pressure of not less than 500 pounds per square inch; (3) the blow-off will be arranged for hand operation, and must have a free opening for discharge, not less than ½ inch in diameter. The drain (discharge valve) must be high enough above the bottom of the trap to prevent being clogged with sediment. All pipe connections to traps will be made with union couplings or by flanges. All traps will have by-pass pipes and valves for convenience of overhauling. All traps will be fitted with valves for testing.

TABLE 1.

Discharge Capacity of Steam Traps.

Make of Trap.	Type.	Nominal size.	W'ght lbs.	Capacity at 250 lbs. pressure.	Remarks.
Vance	Bucket	1-½″	145	26,380	Compound valve.
Ideal	Ball float	1-½″	280	7,040	
Kieley & Mueller	Bucket	1-½″	200	20,250	Compound valve.
Sarco	Expansion	½″	14	2,700	At 200 lb. pressure.
Johns-Manville	Ball float	1-½″	44	3,480	
Armstrong	Inverted bucket	1″	67	3,200	
Anderson	Ball float	1-½″	165	6,230	
Corliss Valve	Ball float	1-½″		4,250	
Swartwout	Bucket	1-½″	145	4,100	
McDaniel	Ball float	1-½″	136	13,800	
Strong	Bucket	1-½″	176	4,800	
Fisher	Bucket	1-½″	151	4,800	
Wright	Ball float	1-½″	109	7,350	Three valves opening consecutively.
Sterling	Ball float	1″	84	3,600	

TABLE 2.

Variation of Capacity with Size of Trap.

Nominal size, Diameter inlet and outlet, inches.	Capacity, pounds per hour under steam pressures of; pound per square inch gage.		
	30	100	200
¾	6,100	4,400	2,400
1	8,600	8,200	6,400
1¼	14,600	9,900	9,100
1½	12,600	13,700	15,500
2	17,600	25,400	25,400

The above capacities were obtained by using an appropriate valve for each steam pressure.

TEST OF MAIN CIRCULATING PUMP OF U. S. S. *NEW MEXICO*.

By J. S. Evans, Commander, U. S. N., Member.

On account of limited facilities at the works of the manufacturer it was not practicable to test the main circulating pumps of the *New Mexico* before installation in the ship and, as the contract specified pumps of a certain capacity against a certain head, it was necessary to test the pumps after installed. As these pumps are motor driven it was possible to measure the power input to the pumping set much more accurately than on any previous test on board ship, this on account of the much greater accuracy of electrical measurements over those obtained from the steam-driven units which had formerly been installed on shipboard. From the tests made on the motors during inspection at the works of the motor manufacturer the efficiency of the motor at various speeds was known and the power input to the pump was thus easily obtained.

The static head was measured at six points as shown in Figure 1, the location of points being as follows:

Point No.	*Location.*	*Height above base line.*
1.	In main injection pipe.....	5 feet 6 inches.
2.	In pump discharge near pumps..	12 feet 3 inches.
3.	At entrance to main condenser..	20 feet 10½ inches.
4.	At middle of rear water chest of condenser	22 feet 6 inches.
5.	At outlet of main condenser....	22 feet 2 inches.
6.	In discharge pipe from condenser	28 feet 5 inches.

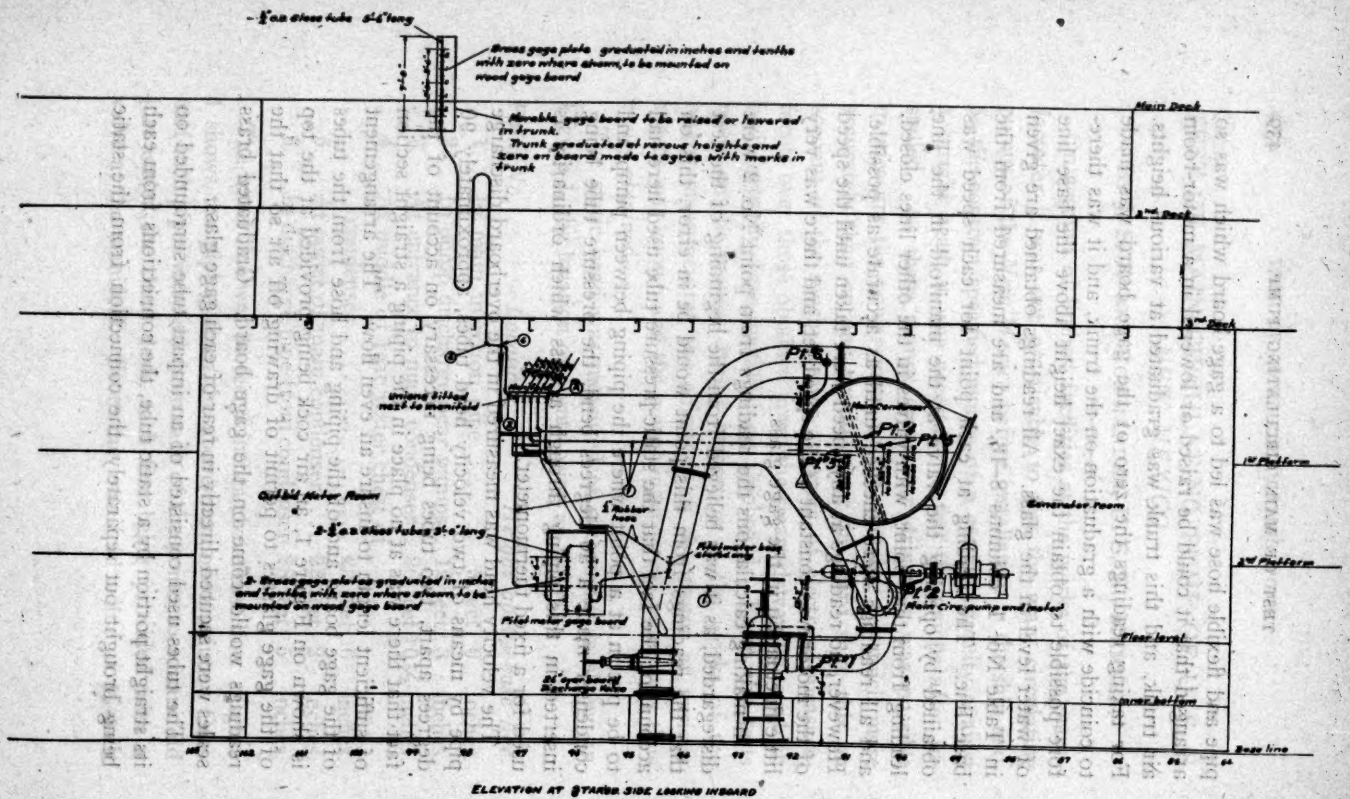
The water line was 30 feet 9 inches above base line. Figure 1 shows the piping from the above-mentioned points. One-half-inch pipes were led from each of the points to a manifold just under the third deck, each line being provided with a valve at the manifold. From this manifold a line of ½-inch

pipe and flexible hose was led to a gage board which was so arranged that it could be raised or lowered in a motor-room air trunk, and this trunk was graduated at various heights. For taking readings the zero of the gage board was made to coincide with a graduation on the trunk, and it was therefore possible to obtain the exact height above the base line of water level in the glass. All readings obtained are given in Table No. 1, columns 8–13, and are measured from the base line. The reading at each point for each speed was obtained by opening the valve at the manifold in the line leading from that point, with valves in the other lines closed, and all readings were checked until as accurate as possible. However, no readings at any speed were taken until the speed of the motor was constant at the speed set, and there was very little fluctuation in the gage glass.

In making calculations the readings from point No. 2 were disregarded, as it was believed from the beginning of the test that the readings from this point would be in error, this on account of the fact that the static-pressure tube used here had to be put in at a point where the piping between pump and condenser made a 90 degrees bend, the pressure tube being inserted in this piping through a boss which ordinarily is used for a fixed thermometer.

The velocity head was measured in the overboard discharge pipe by means of two velocity head tubes, approximately 90 degrees apart, two tubes being necessary on account of the fact that there was at no place in the piping a straight section of sufficient length to insure an even flow. The arrangement of the gage board and of the piping and hose from the tubes is shown on Figure 1, an air cock being provided at the top of the gage glasses to permit of drawing off air so that the readings would come on the gage board. Graduated brass scales were secured directly in rear of each gage glass.

The tubes used consisted of an impact tube surrounded on its straight portion by a static tube, the connections from each being brought out separately, the connection from the static


- ¾" cast brass tube 6'-6" long
Brass gage plate graduated in inches and tenths with zero where shown, to be mounted on wood gage board
Movable gage board to be raised or lowered in trunk.
Trunk graduated at various heights and zero on board made to agree with marks in trunk
Main Deck
2nd Deck
3rd Deck
1st Platform
2nd Platform
Floor level
Inner bottom
Base line
Orlikon Meter Room
Generator Room
Main Condenser
Main Circ. pump and motor
Pitot meter gage board
Pitot meter base stuffing box
¾" Water hose
2-¾" cast brass tubes 3'-0" long
2- Brass gage plates graduated in inches and tenths, with zero where shown, to be mounted on wood gage board
¾" cast brass Discharge tube

ELEVATION AT STARB'D SIDE LOOKING INBOARD

FIG. I.

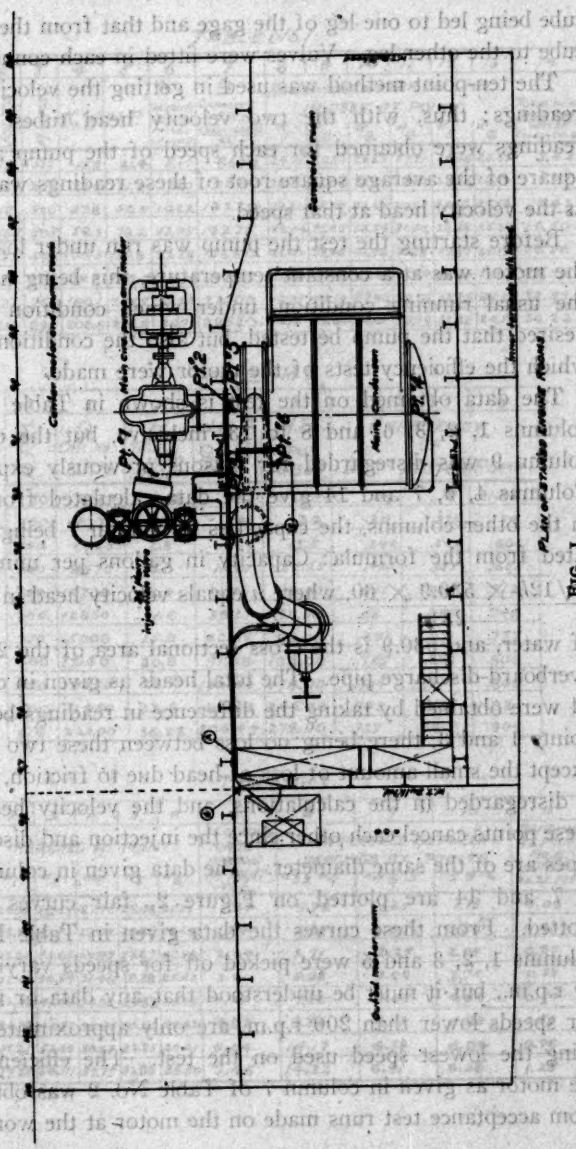


FIG. 1.

tube being led to one leg of the gage and that from the impact tube to the other leg. Valves were fitted in each connection.

The ten-point method was used in getting the velocity head readings; thus, with the two velocity head tubes twenty readings were obtained for each speed of the pump and the square of the average square root of these readings was taken as the velocity head at that speed.

Before starting the test the pump was run under load until the motor was at a constant temperature, this being not only the usual running condition, under which condition it was desired that the pump be tested, but also the condition under which the efficiency tests of the motor were made.

The data obtained on the test is shown in Table No. 1, columns 1, 2, 3, 6, and 8 to 13, inclusive, but the data in column 9 was disregarded for reasons previously explained. Columns 4, 5, 7 and 14 give the data calculated from that in the other columns, the capacities in column 7 being calculated from the formula: Capacity in gallons per minute $= \frac{8\sqrt{12h} \times 530.9 \times 60}{231}$, where h equals velocity head in inches of water, and 530.9 is the cross sectional area of the 26-inch overboard-discharge pipe. The total heads as given in column 14 were obtained by taking the difference in readings between points 1 and 3, there being no loss between these two points except the small amount of loss of head due to friction, which is disregarded in the calculations, and the velocity heads at these points cancel each other since the injection and discharge pipes are of the same diameter. The data given in columns 4, 5, 7 and 14 are plotted on Figure 2, fair curves being plotted. From these curves the data given in Table No. 2, columns 1, 2, 3 and 5 were picked off for speeds varying by 50 r.p.m., but it must be understood that any data or results for speeds lower than 200 r.p.m. are only approximate, that being the lowest speed used on the test. The efficiency of the motor as given in column 7 of Table No. 2 was obtained from acceptance test runs made on the motor at the works of

TABLE No. 1

1	2	3	4	5	6	7	8	9	10	11	12	13	14
R.P.M.	Volts	Amps.	K.W.	H.P.	v Velocity Head in Inches of Water	Q Capacity in G.P.M.	Readings of Water Column in Feet at Point						H Total Head in Feet of Water Calculated
							#1	#2	#3	#4	#5	#6	
200	118	185	16.9	14.4	5.14	2870	24.21	22.26	22.63	27.71	26.46	25.25	6.42
245	118	200	22.6	21.6	7.80	10360	28.71	31.04	31.50	28.63	28.96	28.58	7.78
290	118	320	27.8	26.6	10.22	12220	32.21	32.63	32.63	29.98	27.67	26.58	10.42
335	1155	505	58.3	78.2	13.95	14275	35.04	35.29	36.54	31.46	28.38	28.48	14.50
405	287	440	104.3	139.8	21.28	17680	41.00	39.96	41.96	34.38	29.83	28.79	20.86
440	287	505	119.7	160.4	24.16	18785	44.38	41.46	43.38	35.54	30.63	28.83	23.50
500	287	760	180.1	244.5	31.68	21500	48.21	46.13	47.80	38.63	32.25	26.21	33.59
518	287	880	208.6	278.6	34.72	22520	49.79	47.64	49.01	39.79	32.38	26.50	36.28

TABLE No. 2

1	2	3	4	5	6	7	8
R.P.M.	Q Cap. in G.P.M.	H Total Head in Feet	W.H.P. $\frac{Q \times H}{3955}$	H.P. Input to Set	Efficiency of		
					Set	Motor	Pump
50	2140	0.38	0.2	2.5	.08	.19	.42
100	4275	1.3	1.42	7.0	.20	.375	.54
150	6925	2.8	4.62	13.6	.342	.525	.605
200	8575	5.0	11.01	22.25	.495	.755	.655
250	10700	8.0	21.98	36.00	.611	.858	.712
300	12850	11.6	38.27	58.00	.66	.885	.746
350	15000	15.8	60.85	88.00	.691	.93	.777
400	17150	20.8	91.58	128.00	.715	.938	.806
450	19275	26.4	130.64	178.00	.734	.91	.806
500	21450	33.5	184.48	245.00	.753	.9184	.82
518	22200	36.22	206.44	279.00	.739	.9185	.804

TABLE No. 3

1	2	3	4	5	6	7	8	9	10	11
R.P.M.	Reading at Point					36.28 Feet minus Reading at Point No. 1	Difference between Readings at Points			Reading at Point No. 6 minus 36.28 Feet
	#1	#3	#4	#5	#6		#3 & #4	#4 & #5	#5 & #6	
200	24.21	22.63	27.71	26.46	25.25	1.04	1.92	1.25	1.18	0.98
245	28.71	31.50	28.63	28.96	28.58	1.54	2.87	1.67	1.38	0.33
290	32.21	32.63	29.98	27.67	26.58	2.04	2.71	2.35	2.09	0.93
335	35.04	36.54	31.46	28.38	26.48	2.21	5.08	3.08	2.75	0.39
405	41.00	41.96	34.38	29.83	28.79	4.35	7.58	4.55	4.04	0.54
440	44.38	43.38	35.54	30.63	28.83	4.87	8.34	4.91	4.80	0.58
500	48.21	47.80	38.63	32.25	26.21	6.04	14.17	6.38	6.04	0.96
518	49.79	49.01	39.79	32.38	26.50	7.46	14.22	6.91	6.38	1.25

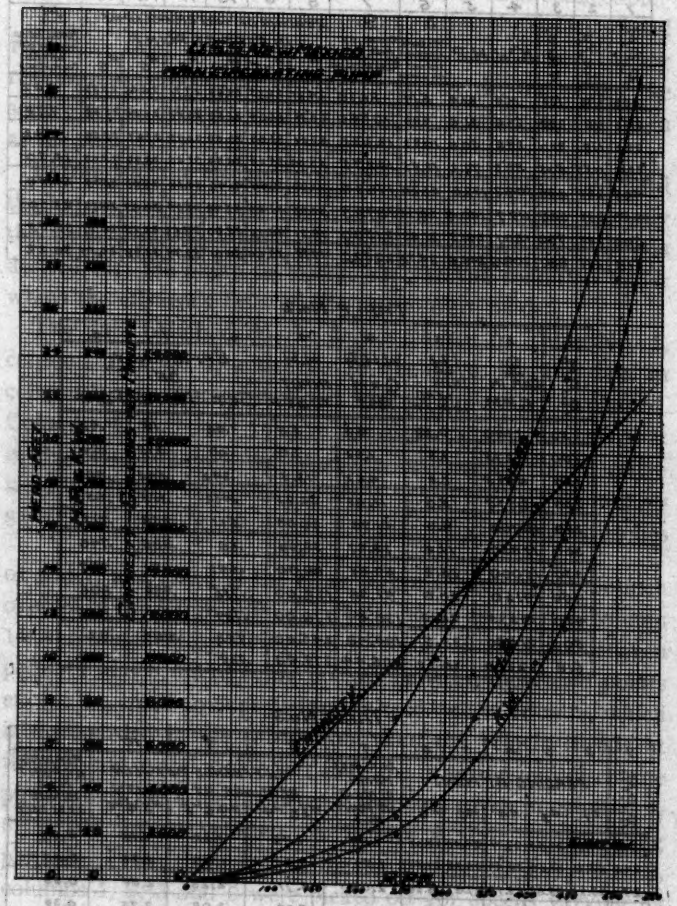


FIG. 2.

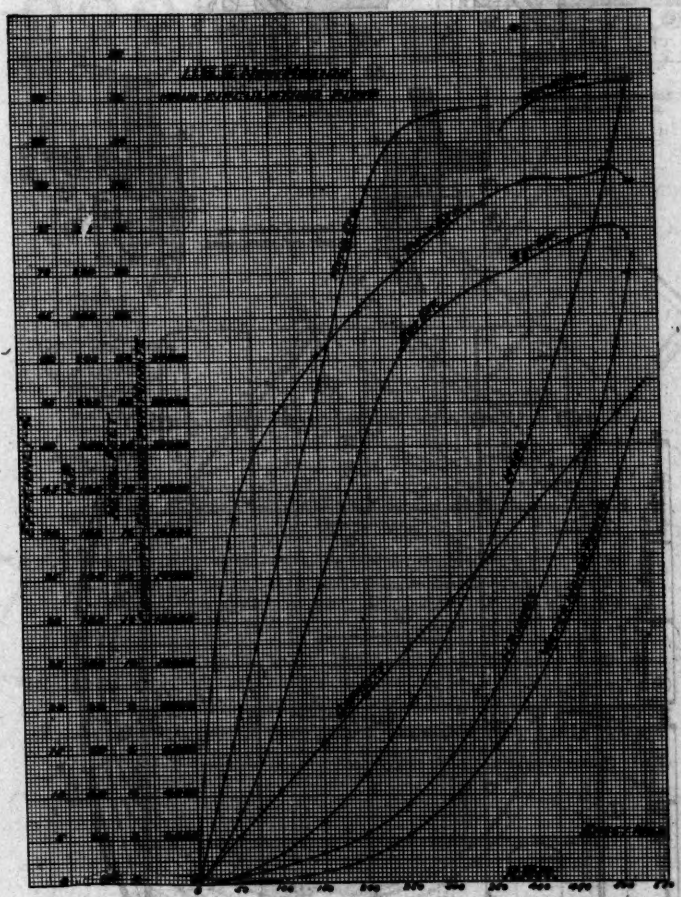


FIG. 3.

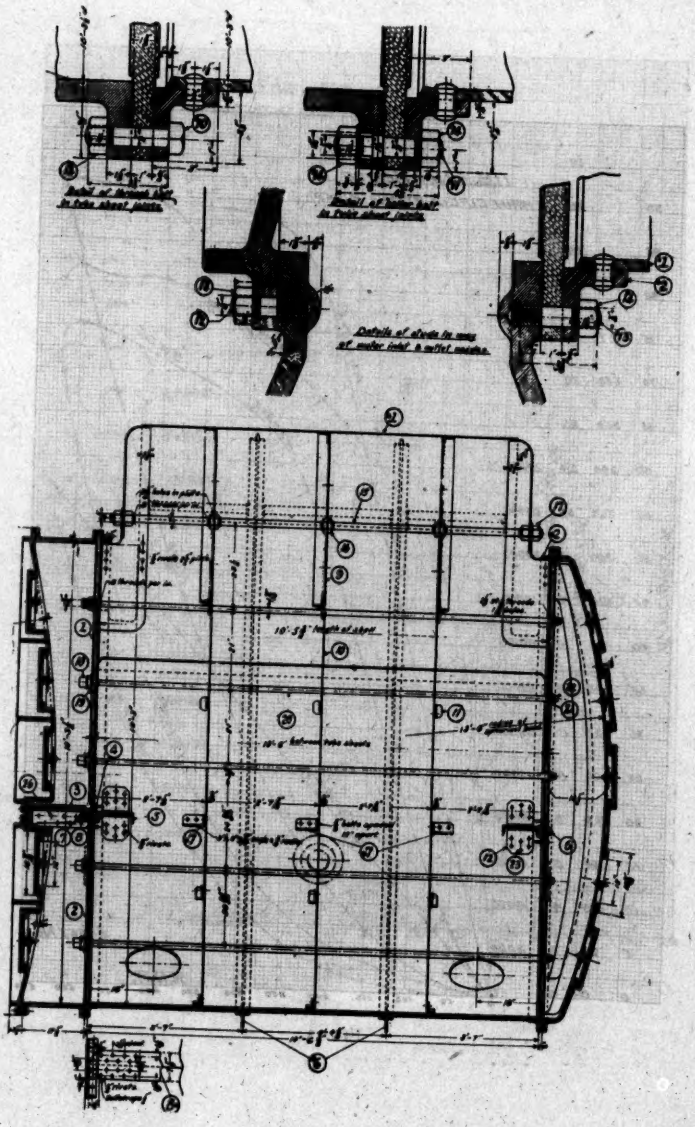


FIG. 4.

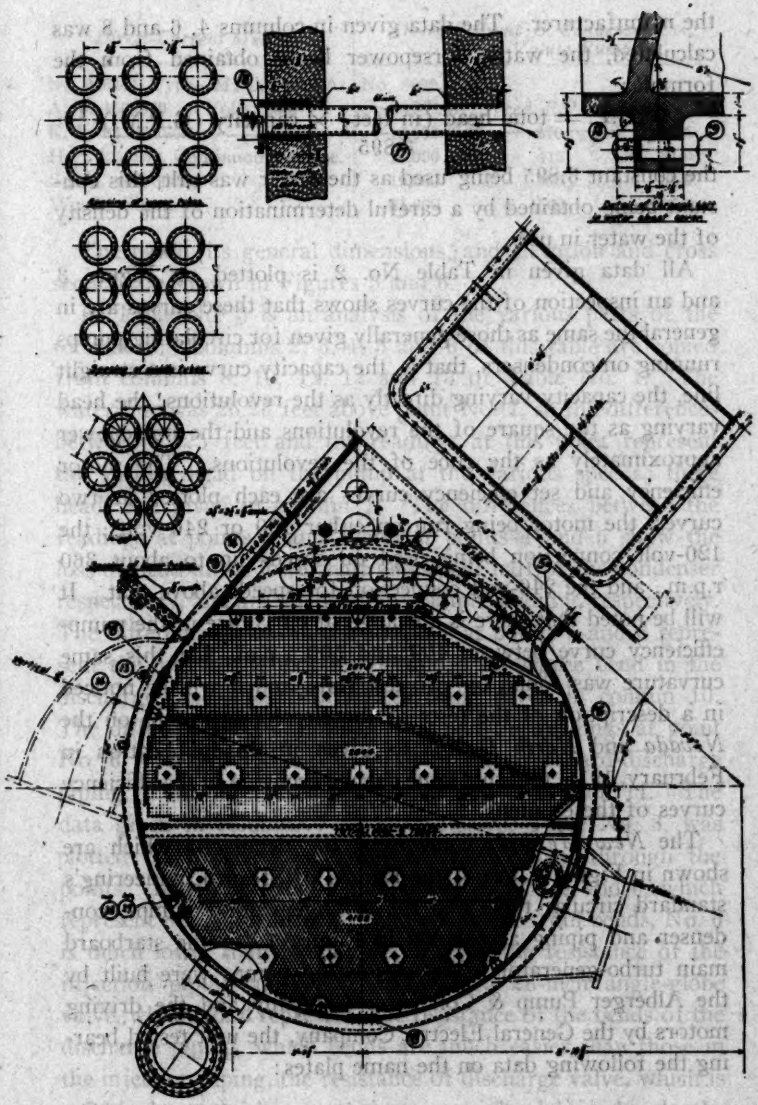


FIG. 4.

the manufacturer. The data given in columns 4, 6 and 8 was calculated, the water horsepower being obtained from the formula:

$$\text{W.H.P.} = \frac{\text{total head (in feet)} \times \text{capacity (G.P.M.)}}{3{,}895}$$

the constant 3,895 being used as the water was salt, this constant being obtained by a careful determination of the density of the water in use.

All data given in Table No. 2 is plotted on Figure 3 and an inspection of the curves shows that these curves are in general the same as those generally given for circulating pumps running on condensers, that is, the capacity curve is a straight line, the capacity varying directly as the revolutions; the head varying as the square of the revolutions and the horsepower approximately as the cube of the revolutions. The motor efficiency and set efficiency curves are each plotted as two curves, the motor being run on either 120 or 240 volts, the 120-volt connection being used on speeds up to about 360 r.p.m., and the 240-volt connection for speeds above that. It will be noted that there is a change of curvature in the pump-efficiency curve between 400 and 500 r.p.m., but this same curvature was found on two test runs, and it will be noticed in a description of the tests of the circulating pumps on the *Nevada* and *North Dakota* as given in this publication in February, 1918, that this same change occurred in the efficiency curves of these pumps.

The *New Mexico's* condensers, the details of which are shown in Figure 4, are of the Bureau of Steam Engineering's standard circular type, and the arrangement of pumps, condenser and piping are shown in Figure 1 for the starboard main turbo-generator. The circulating pumps were built by the Alberger Pump & Condenser Company and the driving motors by the General Electric Company, the unit tested bearing the following data on the name plates:

Direct-Current Motor.	*Alberger Pump & Condenser Co., 24-inch Volute Pump.*		
No. 616244, Type—MPC, Form A.	No. 8692.		
Amp. 450/870 Volts, 230.	*Cap. G.P.M.*	*Speed, R.P.M.*	*Head, Feet.*
K.W. 93.25/186.5.	10,000	270	9.5
H.P. 125/250, Continuous service.	16,000	415	21.0
Speed, 400/500.	19,000	500	30.0
Wound Shunt.	21,600	560	36.0

The pump, its general dimensions, and elevation and cross section are shown in Figures 5 and 6.

In Table No. 3 is an analysis of the various parts of the total head. Columns 2, 3, 4, 5 and 6 of this table are copied from columns 8, 10, 11, 12 and 13 of Table No. 1. The water line was 25.25 feet above point No. 1, so the differences between 25.25 feet and the readings at this point represent the suction head on the pump at the various speeds; these heads are given in column 7. The differences between the readings at points 3 and 4 and at points 4 and 5 show the loss of head in the lower half and upper half of the condenser respectively, and are tabulated in columns 8 and 9, respectively. The differences between the readings at points 5 and 6 represent the loss in the exit nozzle and a 90-degree bend in the discharge pipe, these differences being given in column 10. The differences between 25.25 feet and the readings at point No. 6 represent the loss of head in the overboard-discharge piping, these differences being tabulated in column 11. The data given in columns 7 to 11, inclusive, Table No. 3, was plotted on Figure 7, and curves were drawn through the points. It will be noticed that of curves Nos. 1 and 6, which represent heads due principally to resistance from bends, No. 6 is much lower than No. 1; No. 1 being the resistance of the injection piping and valve which is a 26-inch angle-globe valve, while No. 6 represents the resistance of the bends of the discharge piping, which bends are much easier than those in the injection piping, the resistance of discharge valve, which is a 26-inch gate valve as against an angle-globe valve in the injection line, and the resistance of the water leaving the ship's

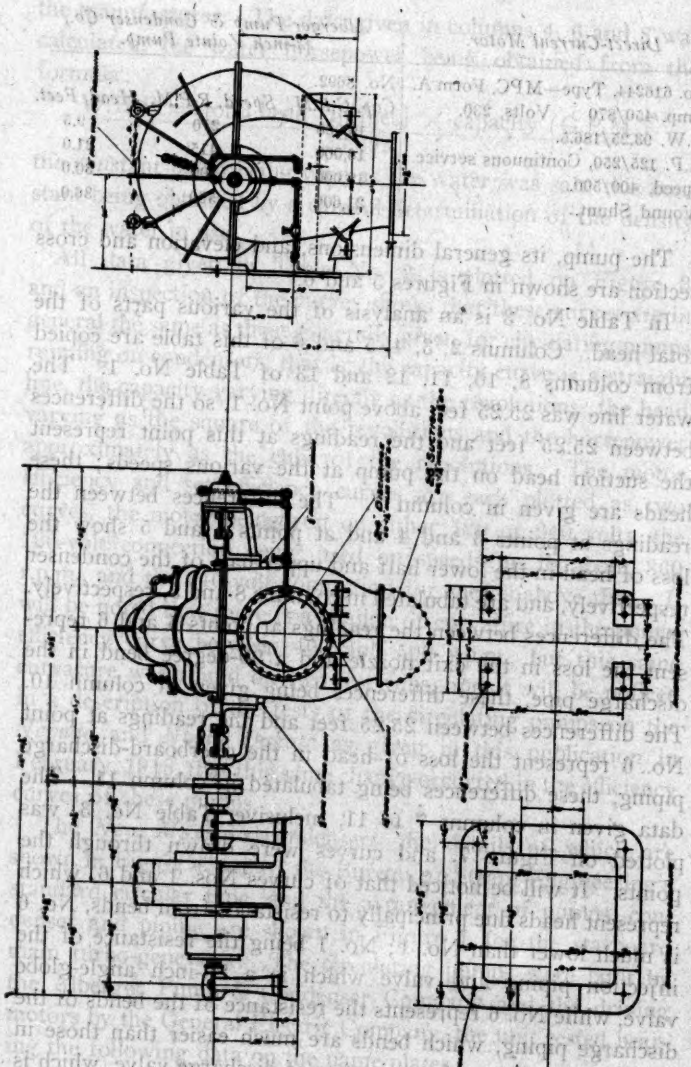


FIG. 5.

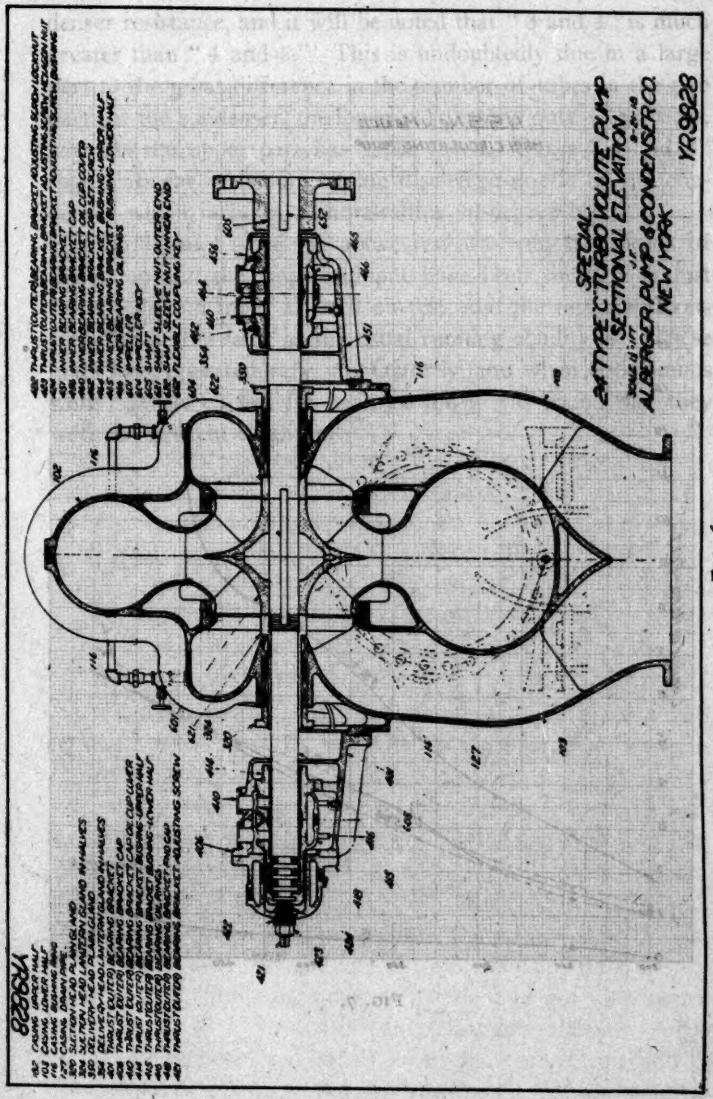

SPECIAL
24" TYPE C TURBO VOLUTE PUMP
SECTIONAL ELEVATION
SCALE 3/4"
ALBERGER PUMP & CONDENSER CO.
NEW YORK
YR9828

FIG. 6.

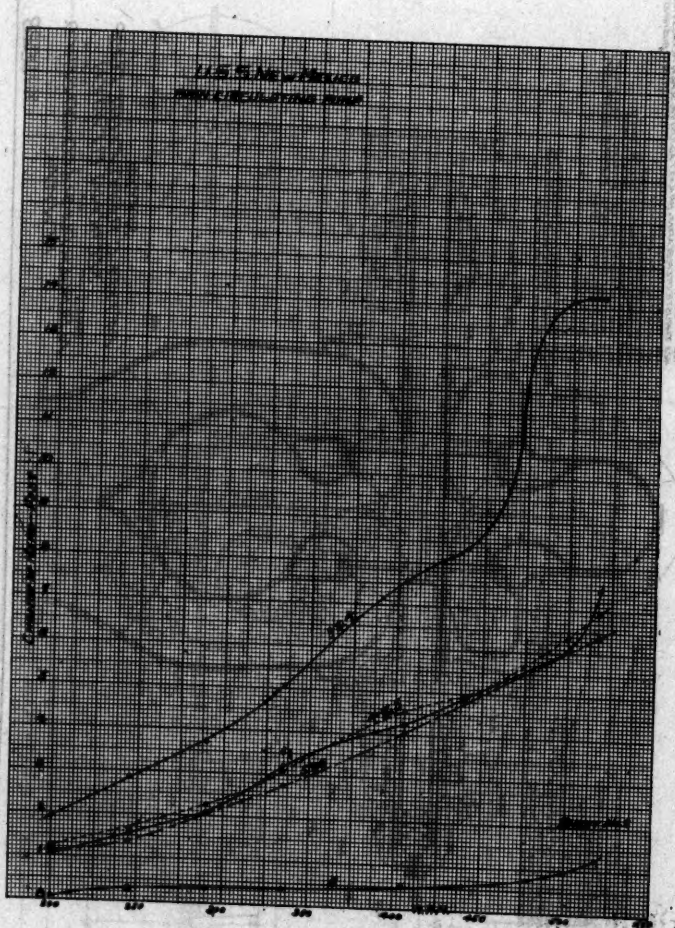


FIG. 7.

bottom. Curves "3 and 4" and "4 and 5" represent condenser resistance, and it will be noted that "3 and 4" is much greater than "4 and 5." This is undoubtedly due in a large part to the great difference in the number of tubes in the two parts of the condenser, the lower part having only 4,182 tubes, whereas the upper part has 4,719 tubes. Curve "5 and 6" represents the resistance of the discharge nozzle of the condenser which includes a short-radius 90-degree bend.

Several times since the above test was run the speeds of the pump and the ampères input have been checked against the speeds and input as used on test, and the test conditions found to be the same as the usual running conditions. These units have operated very satisfactorily, and when both pumps were recently opened for examination it was found that they were in excellent condition.

THE WILLIAMS REFRIGERATING MACHINE.

WM. L. DEBAUFRE.

A test of the Williams refrigerating machine, manufactured by the Electrical Refrigerating Co., Inc., New York City, was made at the Naval Engineering Experiment Station, Annapolis, Md., under authorization of the Bureau of Steam Engineering of the Navy Department. A description of the machine with some of the test results follows:

DESCRIPTION OF MACHINE.

A sectional elevation of the machine appears in Fig. 1 and a sectional plan in Fig. 2. A general view of the machine is shown in Fig. 4 and the compressor parts are shown in Fig. 5.

The operating parts of the machine, including the rotary vapor compressor, the vapor condenser, the refrigerant-control valve, etc., are all inclosed within a vertical cylindrical casing having a dome-shaped head. This casing serves as a reservoir for the refrigerant (ethylchloride) and the lubricant (glycerin), and also forms a jacket around the compressor for the cooling water. The shaft of the compressor projects through a stuffing box in the casing and is driven by an electric motor mounted on a common base plate. Automatic-control devices are mounted on a larger wooden base plate.

The vapor compressor is of the rotary type, as shown in the cross-sectional view, Fig. 1. The rotor, mounted eccentrically within the cylinder, has four radial slots containing blades which are thrown out by centrifugal force against the cylinder. Two pressure-equalizing slots are provided in the forward face of each blade. The various parts of the vapor compressor are also shown in the photograph, Fig. 5, taken after completing the test herein described.

The rotor shaft is supported in ball bearings mounted in the cylinder heads. The heads are bolted to the cylinder with

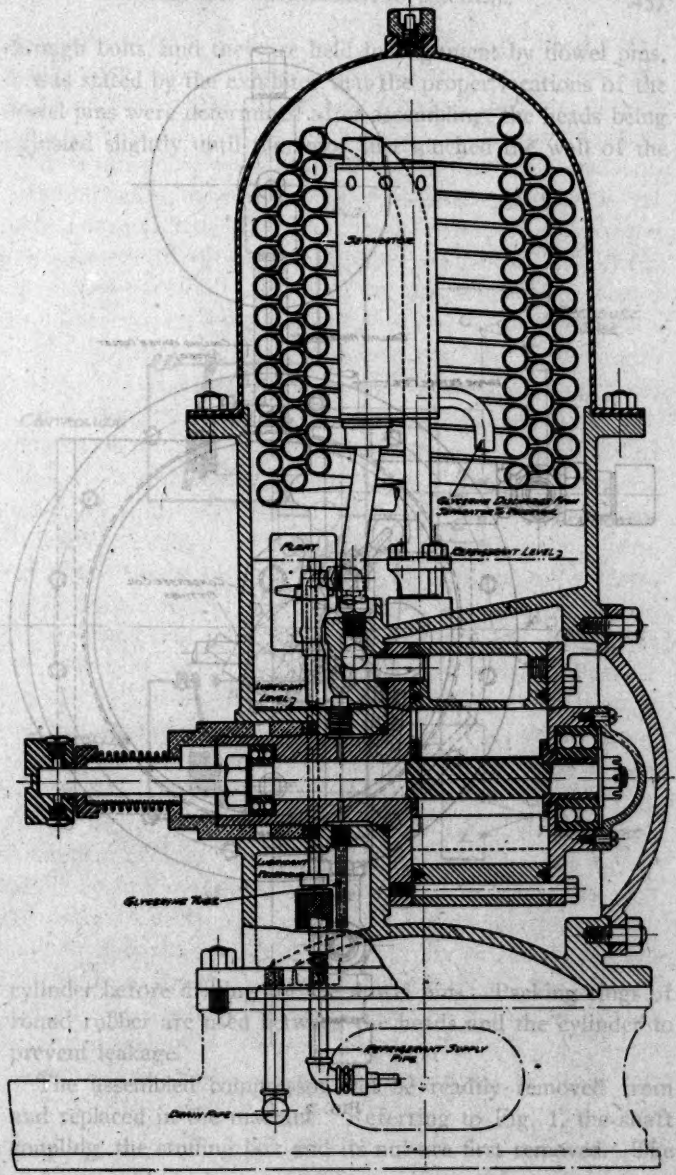


FIG. 1.

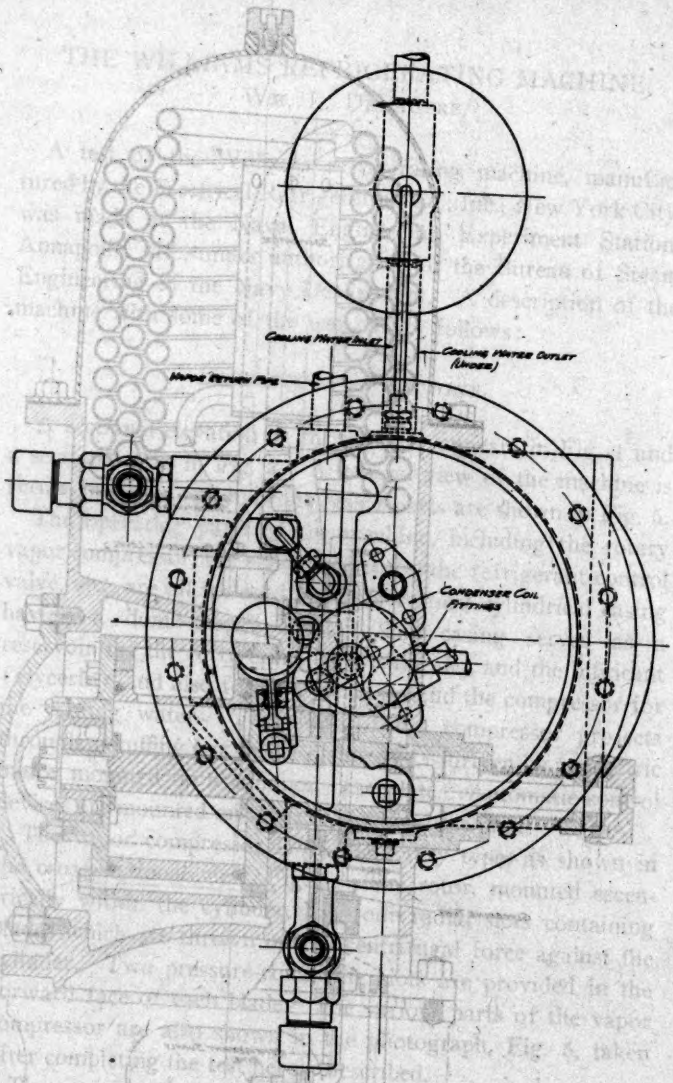

COOLING WATER INLET
COOLING WATER OUTLET (UNDER)
VAPOR RETURN PIPE
CONDENSER COIL FITTINGS

FIG. 2.

through bolts, and they are held in alignment by dowel pins. It was stated by the exhibitor that the proper locations of the dowel pins were determined after assembling, the heads being adjusted slightly until the rotor just touched the wall of the

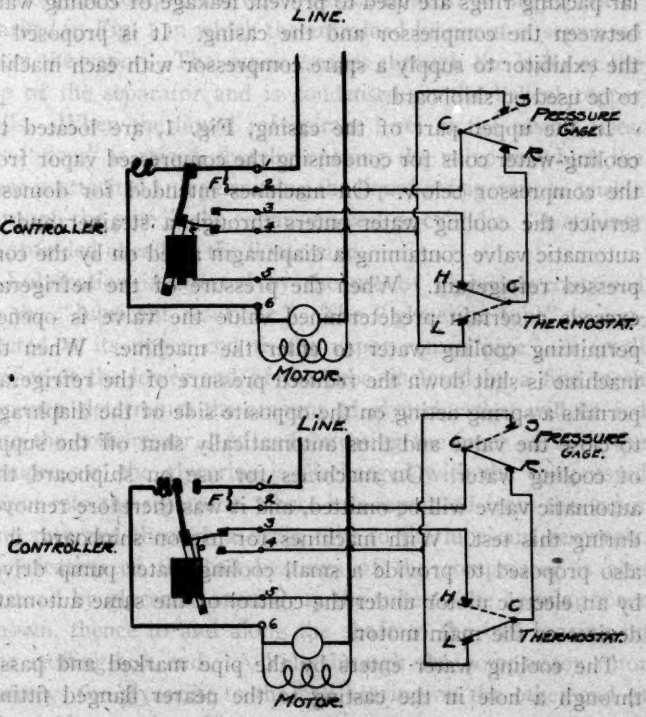


FIG. 3.

cylinder before drilling for the dowel pins. Packing rings of round rubber are used between the heads and the cylinder to prevent leakage.

The assembled compressor can be readily removed from and replaced in the machine. Referring to Fig. 1, the shaft coupling, the stuffing box and its nut are first removed. The

cover on the opposite side of the casing is then taken off and the assembled compressor withdrawn through the opening. The section and discharge connections to the compressor are made by two projecting nipples sealed into corresponding openings in the casing with round rubber packing rings. Similar packing rings are used to prevent leakage of cooling water between the compressor and the casing. It is proposed by the exhibitor to supply a spare compressor with each machine to be used on shipboard.

In the upper part of the casing, Fig. 1, are located the cooling-water coils for condensing the compressed vapor from the compressor below. On machines intended for domestic service the cooling water enters through a strainer and an automatic valve containing a diaphragm acted on by the compressed refrigerant. When the pressure of the refrigerant exceeds a certain predetermined value the valve is opened, permitting cooling water to enter the machine. When the machine is shut down the reduced pressure of the refrigerant permits a spring acting on the opposite side of the diaphragm to close the valve and thus automatically shut off the supply of cooling water. On machines for use on shipboard this automatic valve will be omitted, and it was therefore removed during this test. With machines for use on shipboard, it is also proposed to provide a small cooling water pump driven by an electric motor under the control of the same automatic devices as the main motor.

The cooling water enters by the pipe marked and passes through a hole in the casting to the nearer flanged fitting. The ends of the copper condensing coil are connected to the flanged fittings as marked. After flowing through the condensing coil the cooling water passes through a second hole in the casting and an opening under the plug shown into the space below surrounding the compressor. From this space the water is discharged through the outlet pipe below the inlet pipe.

The refrigerant vapor returns from an expansion coil or cylinder, located where the refrigeration is desired, through a strainer and a check valve to the pipe marked "vapor return" in Fig. 2, and thence directly to the suction side of the rotary compressor. From the discharge side of the compressor, the compressed vapor passes upward to the separator marked in Fig. 1 in which the entrained lubricant is separated from the vapor. The vapor escapes through the holes at the top of the separator and is condensed on the helical copper coils. When the liquid refrigerant level in the casing rises, the "float" is raised, thereby opening the valve to which the float is attached and permitting the liquid refrigerant to trickle through the pipe to the expansion coil or cylinder. A strainer is provided ahead of the float valve.

Lubrication is effected by the use of glycerine contained in the "lubricant reservoir." The "glycerine tube" is connected at its upper end to the vapor return pipe. A small orifice at the lower end of the tube, inclosed by a fine mesh strainer, determines the amount of glycerine that will be fed into the compressor. After lubricating the inner parts of the compressor, the glycerine is discharged with the compressed vapor into the separator, where it is separated from the vapor and returned through a drain tube to the lubricant reservoir. Lubrication of the ball bearings of the compressor is also insured by glycerine passing up the tube to the annular space shown, thence to and along the shaft to the ball bearing on the stuffing-box end. A hole through the compressor rotor conveys the glycerine to the ball bearing on the other end of the shaft.

A specially-designed stuffing box is used. The shaft coupling of case-hardened steel, pinned to the end of the shaft and revolving with it, is counterbored about ⅛ inch around the shaft. A projection on a steel cap is ground to fit into the counterbore of the coupling. This cap is held stationary as the shaft revolves by being connected by riflex tubing to the stationary bonnet. A helical phosphor-bronze spring presses

the cap into the coupling, but the pressure of the glycerine within the folds of the riflex tubing is sufficient to produce a tight joint, and it is evident that the tightness of the joint increases with the tendency to produce leakage by reason of the greater internal pressure.

Sight glasses are provided in the cylindrical casing, as seen in Fig. 4, to show the levels of the refrigerant and the lubricant therein. The casing may be drained of both refrigerant and lubricant through the "drain pipe." Pressure gages are provided to indicate the refrigerant suction and discharge pressures respectively. The latter gage contains a special electrical contact-making device for shutting down the motor should the discharge pressure become excessive.

The refrigerating machine tested was size B–5 and was driven by a 220-volt electric motor at 1,100 r.p.m., operated in conjunction with a thermostat and a pressure gage equipped with an electric contact device.

The automatic operation will be explained by the aid of the diagrams in Fig. 3, the heavy lines showing the main circuit, and the light lines the control circuits. The controller consists essentially of a contact device in the main motor circuit, operated by an electro-magnet which either opens or closes the circuit when energized momentarily. The upper diagram shows the position of the controller when the motor is running, the contact in the thermostat being made on the high-temperature point H, and that in the pressure gage on the running point R. Should the temperature operating the thermostat fall until contact were made on the low-temperature point L, the auxiliary circuit through the electro-magnet on the controller would be closed momentarily; consequently, the main circuit would be opened and the motor and refrigerating machine stopped. The auxiliary circuit would also be closed should the pressure rise until contact were made in the pressure gage on the stopping point S. The auxiliary circuit is always broken on the heavy contacts in the controller and not on the smaller contacts in the pressure gage and thermostat.

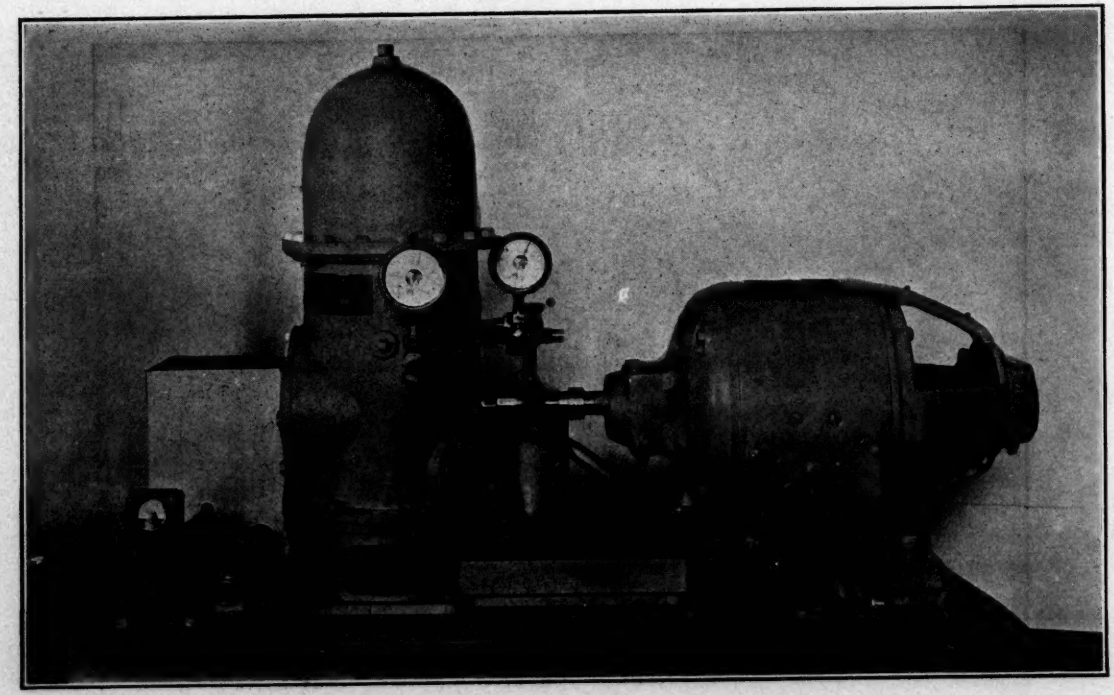


FIG. 4.

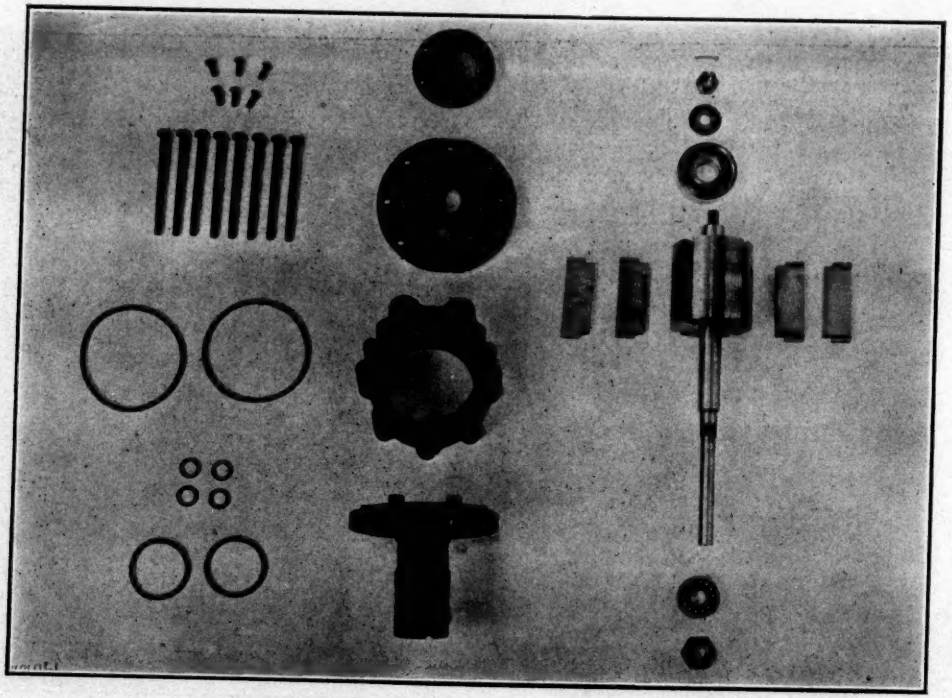


FIG. 5.

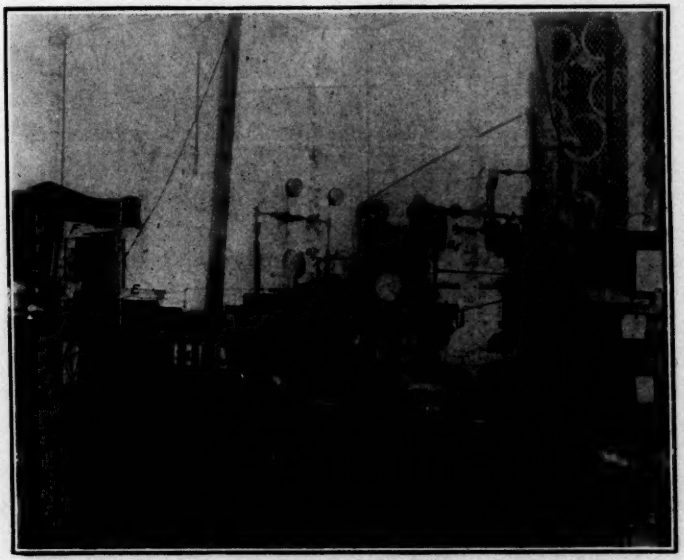


FIG. 6.

FIG. 7.

The lower diagram shows the conditions after the motor has been stopped due to low temperature. If the temperature rises the motor will start again; but should the motor stop by reason of rise in pressure it will not start again automatically when the pressure drops. A push button on the pressure gage must be depressed to again start the motor. A fuse F is provided in the auxiliary-control circuit.

The weight of the machine and motor with the automatic control apparatus mounted on a wooden base plate as shown was 264 pounds. The outside dimensions of the machine were 17½ inches wide by 37¾ inches long by 24½ inches high, equivalent to a volume of 9.3665 cubic feet.

With the refrigerating machine the exhibitor furnished an insulating cabinet, as seen in Fig. 6, containing a brine tank and an expansion cylinder to serve in conducting the test. The cabinet was about 29 inches square by 44 inches high. The brine tank within was 17 inches square by 33 inches deep. The expansion cylinder was 13 inches inside diameter, 14⅜ inches outside diameter and 26½ inches high, the refrigerant being contained in the annular space between the inner and outer walls. The liquid refrigerant supply line of ⅛-inch pipe was connected to the bottom of the expansion cylinder. The vapor line of ⅜-inch pipe branches and was connected to the top of the expansion cylinder at the opposite ends of a diameter. A ⅛-inch glycerin-return pipe, having a valve in it, was also connected from the bottom of the expansion cylinder to the vapor-return line.

OBJECT OF TEST.

The object of the test was to determine the refrigerating capacity and the economy of the machine under various temperatures of refrigeration. Also, to note the general operating characteristics of the machine and the durability of its parts.

TEST ARRANGEMENT AND CONDUCT.

The test arrangement is shown in the photographs, Fig. 6 and Fig. 7.

The electrical input to the motor driving the refrigerating machine was measured by means of an ammeter in the armature circuit, an ammeter in the field circuit, a voltmeter connected across the armature and a voltmeter connected across the series and interpole fields, all mounted on the table seen at the right in Fig. 6. The dial counter D seen in Fig. 7 served to measure the rotational speed of both the motor and the vapor compressor of the refrigerating machine. During the performance runs the automatic temperature control was not in operation. The pressure gage G, however, was connected to the controller to automatically stop the motor should the vapor discharge pressure become excessive.

The test gage G_1 served to measure the pressure of the compressed vapor from the rotary compressor. A mercury thermometer T_1 was inserted in the gage connection to measure the temperature of the compressed vapor. The bulb of this thermometer was directly exposed to the vapor and therefore subject to error by reason of the vapor pressure on the bulb. The thermometer T_2 projected into the vapor space within the condenser and was exposed to radiation to the surrounding cooling coils as well as to the pressure of the compresesd vapor. The thermometer T_3 was tied to the outside of the tube conveying the liquid refrigerant to the expansion cylinder within the cabinet.

The pressure and temperature of the vapor returning from the expansion cylinder were measured by means of the gage G_4 and the thermometer T_4 respectively. The bulb of the latter was directly exposed to whatever vacuum the refrigerant vapor was under, and also to radiation of heat from the pipe walls. The vacuum at the compressor inlet was indicated by gage G_5 and also by mercury column M_1. A mercury trap was provided on the latter to prevent loss of mercury. A

mercury thermometer, T_5, was also inserted in the compressor inlet. The mercury column M_2 measured the drop in pressure in the return vapor line between the expansion cylinder and the vapor compressor, including a check valve and strainer at the refrigerating machine. The thermometer T indicated the room temperature.

The temperature indicated on thermometer T_6, that of the cooling water supplied the machine, was regulated to the desired value by the amount of steam admitted to the surface heater H. The cooling water passed through the coils within the heater. The steam was admitted to the bottom of the shell, which was filled with water, an overflow being provided at the top. In previous tests it had been found that the temperature of the circulating water could be maintained much more nearly constant in this manner than if the steam were admitted in the usual manner and the condensed steam drained off at the bottom. The outlet temperature of the cooling water was measured by thermometer T_7. The thermometer T_8 projected into the cooling water leaving the condenser and entering the jacket space. The pressure drop of the cooling water flowing through the refrigerating machine was indicated on mercury column M_3. The cooling water was discharged into the tanks W and weighed.

The brine in the tank within the cabinet C was circulated by means of the motor-driven pump P. The expansion cylinder within rested on the bottom of the brine tank. The brine from the pump was discharged through a pipe projecting downwards within the expansion cylinder nearly to its base. Rising within the expansion cylinder, the brine overflowed the upper edge and passed downwards over the outer surface of the cylinder to the brine-pump suction pipe connected near the bottom of the brine tank.

The temperature of the brine in the brine-pump suction pipe was measured by the mercury thermometer T_9 and also by a platinum resistance element R connected to the special resist-

ance bridge B, with which was used a highly sensitive galvanometer not seen in the photographs.

A heating coil was also immersed in the brine within the expansion cylinder. The electrical input to this heating coil was measured by the ammeter and voltmeter E. The voltmeter leads were connected to the heating circuit at the cabinet C in order that the instrument should indicate the potential drop within the cabinet only. The watt-hour meter F was also connected in the heating-coil circuit. The aneroid barometer A served to measure the atmospheric pressure.

After setting up the apparatus as described above, the losses in the electric motor were first determined. The armature resistance loss for various armature currents was found by reading the voltage drop across the brushes for armature currents from zero to above the maximum full-load current, the shunt and series fields being disconnected and the armature slowly rotated by hand. The stray power loss was calculated from the armature current and voltage drop at various rotational speeds, obtained by varying the impressed voltage or the field current slightly from the normal value, the motor shaft being disconnetced from the shaft of the vapor compressor.

The platinum resistance element used to measure the temperature of the brine in the cabinet was calibrated after being connected to the bridge, by measuring its resistance in melting ice and also in condensing steam within a special hysometer.

All thermometers used were calibrated before the runs. The pressure gages were calibrated before and after the runs, and at times between the runs. The platform scales were adjusted to read accurately. The electrical instruments had been calibrated before the runs. The resistance thermometer was calibrated as described.

Runs were made at constant speed and with cooling water at room temperature and at 90 degrees F. Runs were also made in which the refrigerating machine was shut down and the valves to the expansion cylinder closed, in order to cal-

culate the heat imparted by the brine pump and absorbed from the room. The results deduced from the test data are plotted in Figs. 9 to 11 inclusive.

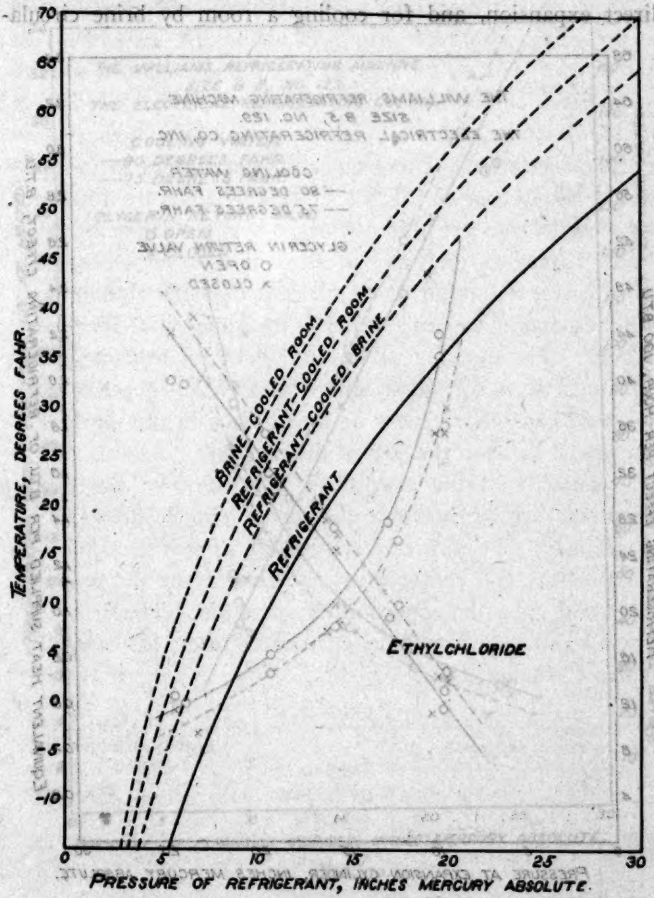


FIG. 8.

The saturated vapor temperatures of the refrigerant, ethylchloride, are taken from the full-line curve of Fig. 8, corresponding to the absolute pressures noted. In this figure are

also plotted dotted lines ten, fifteen and twenty degrees above the full-line curve. These temperature differences are considered fair values for cooling brine, for cooling a room by direct expansion, and for cooling a room by brine circula-

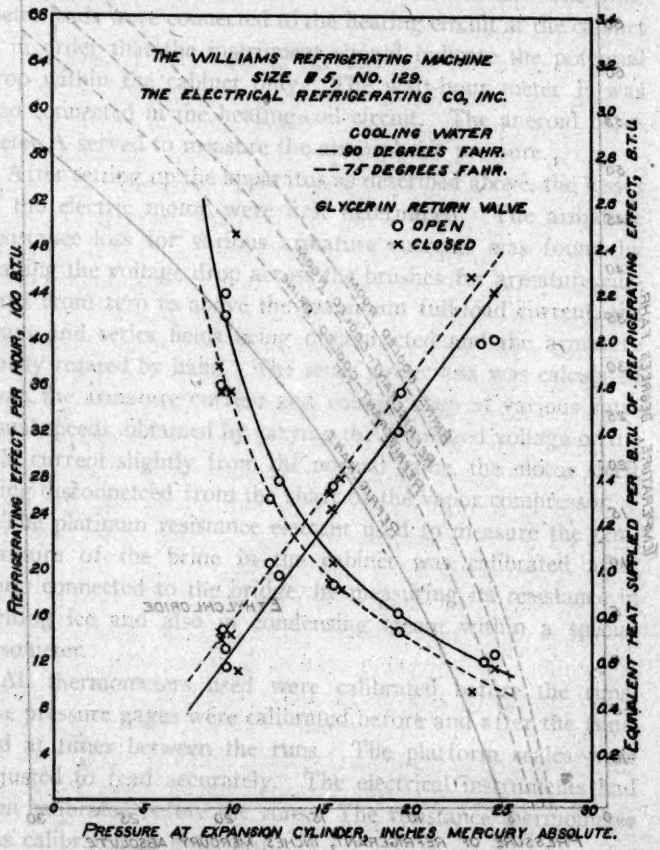


FIG. 9.

tion, respectively. In the case of a certain room which is to be kept at 40 degrees F., the vapor pressure carried in the expansion coils must be 13.8 inches mercury absolute if the room is cooled by brine circulation. If cooled by direct expansion,

the vapor pressure must be 15.4 inches mercury absolute. To cool brine to 40 degrees F., the vapor pressure would be 17.5 inches mercury absolute. More generous provision of sur-

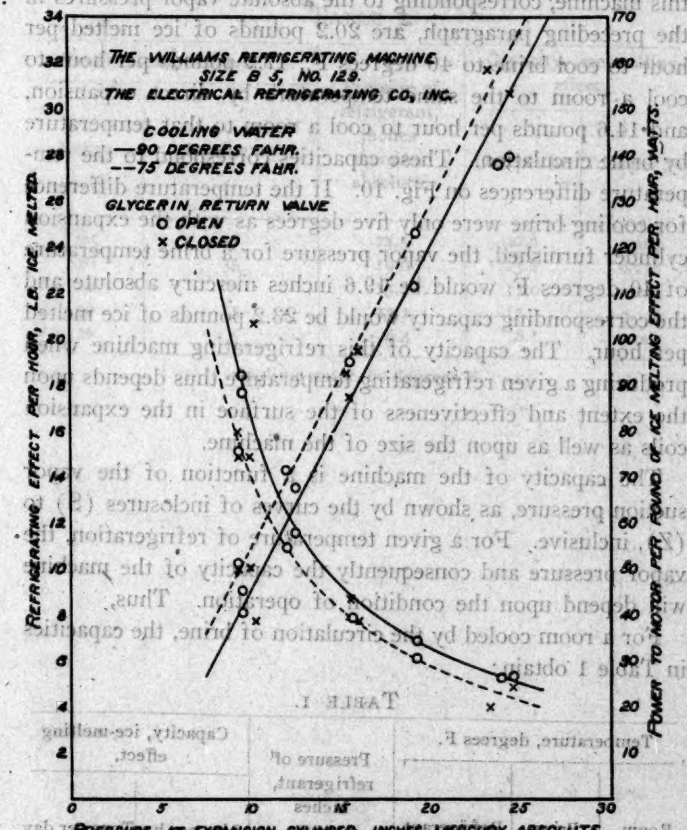


FIG. 10.

face in the expansion coils would, of course, result in lower temperature differences than indicated in Fig. 10. Thus, with the expansion cylinder furnished for this test, the temperature difference was only about five degrees F.

At whatever vapor pressure it may be necessary to operate

the machine tested, the corresponding capacity and economy are shown by the curves of Figs. 9 and 10. We find that with cooling water at 90 degrees F., the refrigerating capacities of this machine, corresponding to the absolute vapor pressures in the preceding paragraph, are 20.2 pounds of ice melted per hour to cool brine to 40 degrees F., 17.2 pounds per hour to cool a room to the same temperature by direct expansion, and 14.6 pounds per hour to cool a room to that temperature by brine circulation. These capacities correspond to the temperature differences on Fig. 10. If the temperature difference for cooling brine were only five degrees as with the expansion cylinder furnished, the vapor pressure for a brine temperature of 40 degrees F. would be 19.6 inches mercury absolute and the corresponding capacity would be 23.2 pounds of ice melted per hour. The capacity of this refrigerating machine when producing a given refrigerating temperature thus depends upon the extent and effectiveness of the surface in the expansion coils as well as upon the size of the machine.

The capacity of the machine is a function of the vapor suction pressure, as shown by the curves of inclosures (S) to (Z), inclusive. For a given temperature of refrigeration, the vapor pressure and consequently the capacity of the machine will depend upon the condition of operation. Thus,

For a room cooled by the circulation of brine, the capacities in Table 1 obtain:

TABLE I.

Temperature, degrees F.			Pressure of refrigerant, inches mercury absolute.	Capacity, ice-melting effect.	
Room	Brine	Refrigerant		Lbs. per hr.	Tons per day
60	50	40	21.9	26.7	0.321
50	40	30	17.5	20.2	0.242
40	30	20	13.8	14.6	0.175
30	20	10	10.8	10.2	0.122
20	10	0	8.3	6.4	0.077

Cooling water, 90 degrees F.

For a room cooled by direct expansion, the capacities corresponding to various temperatures are given in Table 2.

TABLE 2.

Temperature, degrees F.		Pressure of refrigerant, inches mercury absolute.	Capacity, ice-melting effect.	
Room	Refrigerant		Lbs. per hr.	Tons per day
60	45	24.5	30.6	0.367
50	35	19.6	23.2	0.278
40	25	15.4	17.0	0.204
30	15	12.2	12.2	0.145
20	5	9.4	8.1	0 097

Cooling water, 90 degrees F.

PROBLEM CORNER.

As an additional feature of the JOURNAL, friends of the Society have suggested that a few pages be devoted to the proposition of engineering problems and their solutions, the object being to stimulate interest in reviewing mathematical subjects and to establish a clearing house for the solution of engineering problems encountered in practice.

Such a feature, if successful, will be of value to the Society and the JOURNAL, in that it will provide a means for proposing problems whose solutions are of general interest and may well be the premises for discussion at length in the form of contributed articles.

To be successful this feature must arouse the interest and appreciation of the members of the Society and the subscribers to the JOURNAL. The principles involved in the problems proposed must be varied and the solutions must range in difficulty from simplicity to the latest methods of higher analysis developed for the treatment of practical problems in radio-telegraphy, elastic structures and thermodynamics.

Comment and criticism from the readers of the JOURNAL will be appreciated. If the problems proposed in this issue of the JOURNAL and the idea of incorporating similar problems in future issues meet with favor, effort will be made to publish as interesting and out-of-the-ordinary problems as can be obtained. Readers are invited to submit problems in which they are interested. Solutions will be published if of general interest and space permits.

PROBLEMS FOR SOLUTION.

Problem No. 1—Administration—An Actual War Problem.

In October, 1918, the Bureau of Steam Engineering received the following despatch from Admiral Sims:

"U. S. S. ———, after discharging completed, will require docking to renew broken stern bush. In order to reduce time

in dock, request you wire dimensions of brass stern bush as fitted in stern tube in order rough casting may be made before vessel docks. This information can not be obtained from vessel prior to docking."

The vessel referred to had been built on the Pacific Coast, and the only blue print showing dimensions of the stern-tube bushing was in the files of the Emergency Fleet Corporation in Philadelphia. A description of the stern-tube bushing with all dimensions was obtained by telephone and the following despatch transmitted:

"Outboard stern-tube bushing sixty and one-half inches overall comma outside diameter nineteen and one-quarter comma inside diameter eighteen comma outboard flange diameter twenty-three comma thickness one and one-quarter comma inboard keeper flange sixteen and one-half inside diameter thickness one-half period Inboard bushing twenty-four and one-half inches overall comma all diameters same as outboard bushing comma bulkhead flange twenty-three diameter thickness one and one-eighth comma set back three-quarters from inboard end period wood one inch thick period Shaft diameter over sleeve fifteen and fifteen-sixteenths comma bearing sleeve sixty-three and one-quarter inches long period All dimensions finished."

Referring to this case some time later, the Bureau was advised that "the cable in question came without error; the rough bushing was cast from the dimensions given, but not machined until the vessel was docked and finished dimensions checked. The results are very gratifying in that the casting was available on the day the ship docked, thus reducing the stay of the vessel in dry dock by four days."

Make a working sketch of the stern-tube bushings from which the foundry and machine shop work can be done.

Problem No. 2—Design—Mechanics—Theory of Elasticity.

In heavily loaded frame structures the elongation or compression of the various elements produces secondary stresses

in the elements which may be sufficiently great to cause failure of the entire structure—the Quebec Bridge disaster was claimed to have been so caused. To safeguard frame structures architects and civil engineers make use of the "principle of least work" in assigning safe stresses to the structural elements. The "principle of least work" states that the sum of the elements of work done in distorting the individual structural elements is a minimum for any given load on the entire structure. In reduction gearing for marine turbines the pinions transmit a considerable torque and undoubtedly suffer some torsional deflection, but for each half of each pinion to transmit the same tooth pressure there must be no relative torsional deflection between the two halves of the pinions. The Melville-McAlpine reduction gear recognizes this principle and provides for axial deflection of the floating frame.

Can the "principle of least work" be utilized in the design of pinion shafts or couplings so that the tooth pressures on each half of the pinion will always be equal? How?

Problem No. 3—Design—Mechanics—Calculus—Shop Practice.

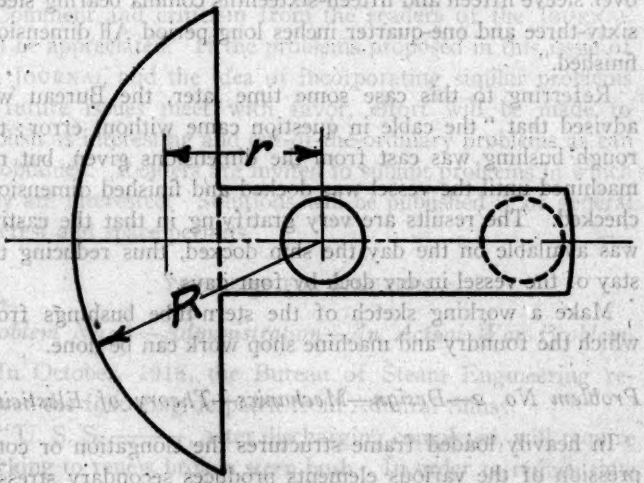


FIG. 1.

A designer, in laying out a counterweight for a crank shaft, plans to give the counterweight the shape of a circular segment the outside radius of which, R, is given. How can such a counterweight be designed so that its axial width may be minimum? This means, of course, that the product of the area of the segment by the distance, r, of its center of gravity from the center must be maximum. Having found the solution, indicate also the curve, giving the efficacy of such a counterweight between the two extremes, $r = R$ and $r = o$, the result is perfectly simple and scientifically correct, but the designer does not like it and decides to sacrifice some of the features to secure a result more attractive from the standpoint of the practical engineer. By what considerations will he be guided in the contemplated departure from the exact solution?

Problem No. 4—Mechanics—Calculus.

A rod, 4.9 feet long, suspended by one of its ends, will perform small oscillations at the rate of 60 single beats per minute. It is also a well-known fact that, if suspended by a point, one-third the length below the top end, it will oscillate with the same frequency; while, when suspended at its middle, such a pendulum will, of course, have zero frequency.

Consequently there must be an intermediate point, such that the corresponding rate will be a maximum.

Required: (a) to locate such a point;
(b) to find the corresponding rate;
(c) to draw a curve connecting the rate with the corresponding point of suspension.

(This is understood to have been proposed originally by Admiral G. H. Burd.)

Problem No. 5—Mechanics—Calculus.

Find the strongest beam of rectangular section that can be sawed out of a round log whose diameter is D. This, of course, is a well-known problem, but it can be solved in two ways; one involves trigonometry and the other does not. Both can be obtained in two or three lines; having secured these,

draw a curve, of which the solution just found will be the maximum, and whose extremes correspond to the zero-breadth and zero-height conditions.

Problem No. 6—Mechanics—Calculus.

Find the stiffest beam of rectangular section that can be sawed out of a round log whose diameter is D.

Problem No. 7—Mechanics—Calculus.

Imagine a coil spring of the following characteristics: the extension of one inch means the pull of λ pounds; then λ is the so-called spring constant. To draw a diagram of such a spring assume that one end is fastened; take the other end as origin of coördinates and direct the axis of abscissae along the center line of the spring; the ordinates will represent the pull, and the deflection curve will, of course, be given by a straight line, beginning at the origin and inclined to the axis of abscissae at an angle such that its tangent λ.

It is required:

(a) To complete the diagram for the case when the spring is to work in compression as well as in tension.

(b) To draw a diagram for the case where a flat plate is placed between two springs, originally free, of characteristics λ and μ, respectively.

(c) Same as under (b), except that the springs are initially loaded, so that the flat plate is subject to a compression of p pounds on each side.

(d) Same as (b), except that inside each spring are placed shorter springs, of characteristics λ and μ, respectively; the latter springs do not come into play for deflections smaller than m on the right and n on the left side.

(e) Same as (c) and (d) combined.

Problem No. 8—Mechanics—Calculus.

Find general formulae of work (or energy), W, in terms of displacement, e, corresponding to the five types given in problem No. 7.

Find the displacement rate of energy dw/dl, and study the law of increase of the energy if the independent variable, l, is equicrescent.

Find the energy rate of displacement, dl/dw, and show the law of increase of the displacement, in each case, if the independent variable, W, is equicrescent.

Problem No. 9—Mechanics—Calculus.

A weight, P pounds, is supported by a coil spring, free length, l, characteristic, λ; find the period of free oscillation of such a system.

Problem No. 10—Mechanics—Calculus.

A weaker spring, length l_1, characteristic μ, is placed inside the spring of problem No. 9. Find the free period of the same weight, P.

Problem No. 11—Mechanics—Calculus.

If a half-inch hole is drilled through the shaft of a motor at 45 degrees to its axis, and if the end of a piece of cold-rolled steel 12 inches long is inserted and fastened therein, is such a system out of balance statically, dynamically or both? The system is started from rest and accelerated uniformly for t seconds, at the end of which period the speed is reached of n r.p.m., which thereafter remains constant. At what time is the rod in question subject to greatest stress, and what is the horsepower required in bringing the motor to that speed?

Problem No. 12—For Amusement—Algebra.

While not an engineering problem, the following is of historical interest:

"The combined ages of Mary and Ann are forty-four years, and Mary is twice as old as Ann was when Mary was one-half as old as Ann will be when Ann is three times as old as Mary was when Mary was three times as old as Ann. How old is Ann?"

NOTES.

NAVAL CONSTRUCTION (BRITISH) DURING THE WAR.

By Sir Eustace Tennyson d'Eyncourt, K.C.B., Vice-President.*

Having obtained the sanction of the Admiralty to give some particulars of vessels added to the fleet during the war, I suggested to the Council of the Institution when putting forward proposals for reading a paper, that in view of the very large amount of design work and construction carried out by the Admiralty during the period under review, it would, on the whole, be most useful and interesting to give a sketch and general summary of all the work rather than select a few types of warships and give a more detailed account of their design and construction.

I need hardly say that within the limits of time and space available for a paper read at this meeting it would be impossible to go fully into more than one or two of the most important designs, although perhaps it may be thought that a complete story of some of them, such as the *Renown* and *Repulse*, or the monitors, might make more attractive reading to our members. Still as the accounts of so many other important types would have had to stand over it appeared that the best course would be, as I have said, to give a general account of as many as possible of the ships added to the fleet during the last four and a half years, and this I will endeavor to do.

Immediately after war was declared, great pressure was exercised to complete the ships then building for the Navy, and to order such other vessels as could be designed and finished in the shortest possible time. Many very wise people told us the war would only last a year or possibly eighteen months—in fact that it could not be carried on for a longer period.

This view necessarily colored all that was done in the way of naval design and construction. Generally speaking, therefore, the construction of new battleships was ruled out. With the acquistion of the *Agincourt*, *Erin* and *Canada*, which were building here for foreign Governments in private yards, and bearing in mind the early completion of the remaining two vessels of the "*Iron Duke*" class, shortly to be followed by the vessels of the "*Queen Elizabeth*" class, we had a great preponderance of heavier capital ships, or Dreadnaughts, over the enemy, and as this class of ship takes longer to design and construct than any other, it was obviously a prudent course to concentrate on such types as were specially needed and could be built more quickly.

It should also be remembered that the menace of the submarine, which was from the first beginning to loom as a vital factor in the war, pointed in the direction of large numbers of patrol boats, torpedo-boat destroyers, and smaller types of vessels to deal with this menace. No time, therefore, was lost in placing orders for additional destroyers, submarines, light cruisers, sloops, mine-sweepers, patrol boats, &c.; and it very soon became clear that the dockyards and the regular warship-building contractors would not be able to cope with the mass of new construction that was required.

Accordingly, orders for many of the last-named classes were placed with builders who had hitherto only been accustomed to mercantile work.

* Paper read before the Institution of Naval Architects, April 9, 1919.

With the arrangements that were made, however, for superintending and overseeing the work by the Admiralty, with the assistance of the registration societies—Lloyd's and the British Corporation—very little difficulty was experienced in getting the work satisfactorily carried out by the firms new to this class of shipbuilding, and I think the results show what success attended the arrangements made.

To take ships added to the Navy during the war in the proper order, it is necessary to begin with battleships of the "*Iron Duke*" class. The particulars of all previous Dreadnaughts are pretty well known, and have been published. The "*Iron Duke*" class, of which there were four, followed the "*King George V*" class both in sequence of time and in general characteristics. The same main armament, similarly arranged, with the five turrets all on the center line of the ship, was adhered to, the chief difference in the "*Iron Dukes*" being that instead of the 4-inch guns forming the secondary armament, a battery of twelve 6-inch guns protected by 6-inch armor was, after considerable discussion, finally decided upon. The protection also was somewhat increased over that of the *King George V*, involving an increase in dimensions over any of our previous battleships, due to the addition of these weights and of other items. Two of the class had been laid down in January, 1912, and two in May, the four vessels being completed in March, June, October and November, 1914, so that two were ready just before, and two shortly after, the declaration of war. Four torpedo tubes were carried in lieu of three in the previous ships, and after the Battle of Jutland, a considerable amount of additional protection was added over the magazines—a course which was practically adopted in all our ships at that time as a precautionary measure. Only in one case was any portion of a shell found to have penetrated below the protective deck; but with the ever-increasing range at which actions have been fought, and the increasing penetration of improved shell, the danger of the decks being inadequate had to be considered.

The tables herewith give general particulars of these ships and of all the others with which I am dealing in this paper. It should be mentioned, however, that the speed obtained on trial was approximately 22 knots, or about a knot in excess of the legend speed.

Special interest is attached to this class, as the *Iron Duke* was the fleet flagship during the whole time of Admiral Jellicoe's appointment as Commander-in-Chief, and she was in action at Jutland with her sister ships. The *Marlborough*, it should be specially noted, was the only British battleship of the post-Dreadnaught type struck by a torpedo during the whole war, and the value of the longitudinal protective bulkhead and of the subdivision and arrangements adopted was clearly shown, as the ship was able to remain in the line, no vital damage being done. She was afterwards safely docked in the Tyne and repaired. This is, I think, specially interesting, as many of our older ships, some with center-line bulkheads and with other arrangements not so good for dealing with under-water damage, were sunk in the Dardanelles and elsewhere by enemy torpedoes.

The next type to note is the "*Queen Elizabeth*" class of the 1912-13 program. Three of these vessels, after taking a little more than two years to build, were completed in January, March and October, 1915. The other two were completed in February, 1916. A very considerable departure was made in the *Queen Elizabeths* from any previous Dreadnaughts, the 15-inch gun taking the place of the 13.5-inch and the designed speed being increased by 4 knots over our previous Dreadnaughts, whilst the secondary armament was similar to that of the *Iron Dukes*, consisting of 6-inch guns. Their very great increase of speed involved practically doubling the horsepower necessary to give the 25 knots desired, and the

great increase in the weight of the 15-inch guns and mountings over the 13.5-inch meant accepting only four turrets with eight 15-inch guns, as against five turrets with ten 13.5-inch guns in our previous ships, and even so, the armament was considerably heavier. The further great departure from previous practice in battleships was the adoption of oil only as the fuel. This necessitated special arrangements of the oil bunkers, many of which were 30 feet in height, and required special construction to withstand the head of oil. The armor and protection were fully maintained as compared with previous ships, but all these additions involved increasing the displacement of 27,500 tons.

In the Battle of Jutland, the Fifth Battle Squadron, consisting of four vessels of this class, were heavily engaged for several hours, and although they inflicted and sustained heavy punishment, especially in the case of the *Warspite*, all the vessels gave a splendid account of themselves and were not seriously damaged or put out of action. After the Battle of Jutland, additional protection was added to the magazines. It may be mentioned that the oil fuel proved a complete success, it being found easier to keep up a high sustained speed, and a smaller complement also is, of course, involved, as there is great saving in *personnel* as against that required for a coal ship.

I should mention that Sir Philip Watts was responsible for the design of the "*Iron Duke*" and "*Queen Elizabeth*" classes, thus completing a series of 27 battleships of the Dreadnaught class designed and built during his tenure of office at the Admiralty—in addition to the large number of battle cruisers and light cruisers and other vessels built during that period—truly a great record.

BATTLESHIPS OF THE "ROYAL SOVEREIGN" CLASS.

The next in order came the "*Royal Sovereign*" class of the 1913-14 program. These vessels were to have the same armament as the *Queen Elizabeth*, but as there was some question about the supply of oil fuel when the design was discussed, it was decided to revert to coal, and also to accept the slower speed of 21 knots, which would make them more homogeneous with other Dreadnaughts. Subsequently, when the vessels were in process of construction and the great advantages of the use of oil fuel with other types of warships became apparent, it was decided to change from coal to oil, and it was anticipated that increased power giving a speed of about 23 knots would be obtained. As a matter of fact when fully laden with about 4,000 tons of oil, the *Revenge* attained 22 knots, which was equal to about 23 knots in the designed load condition. A different disposition of deck and side armor was also adopted by which the thick protective deck at the center of the ship was brought up to the level of the main deck; this portion of the protective deck being thus well above the level of the deep load-line, and giving more protected freeboard in the damaged condition than in any of our earlier battleships. This was an important feature, a somewhat reduced metacentric height was decided upon for these ships with a view to making them steadier gun platforms than some of the ships with larger G.M. The vessel was provided with good under-water protection, which in certain of the ships was further reinforced by adding outside bulge protection. This was done to the *Ramillies* before her launch and also to two other vessels of the class after they had been in commission some time, during refit, and it is proposed to add the bulge to the remaining two ships of the class when opportunity offers. The addition of bulges was suggested by myself originally for the "*Edgar*" class for which I designed this form of protection in 1914, after considerable experiments had been made. The results have proved the efficiency of the bulges. Further particulars of the "*Royal Sovereign*" class can be seen from Fig. 1 and the table.

BATTLESHIPS TAKEN OVER FROM FOREIGN GOVERNMENTS.

The three ships taken over from foreign Governments were of different types, as shown in the table of particulars and Figs. 2 and 3.

H. M. S. *Agincourt* (Fig. 2) was commenced in September, 1911, for the Brazilian Government, when I took out various designs (got out under Mr. Perrett, at Elswick) to Rio de Janeiro, and finally settled on the design of the *Agincourt*, after modifying it very considerably on the spot. The Brazilian authorities, after much discussion, decided upon fourteen 12-inch guns, twin-mounted in seven turrets. This involved a ship with a length of 632 feet between perpendiculars and 671 feet 6 inches over all. The main armor was somewhat lighter than our British Dreadnaughts (as seen from the particulars opposite), and in other respects, such as fueling facilities, the ship hardly came up to the British standard. However, she was well reported on and the 14 big guns were liked by the gunnery officers, who preferred a large number of guns for their salvoes. Certain alterations had to be made to fit her for our service, but in the main she was left as designed. I should perhaps mention that in 1914 she was transferred from the Brazilian Government to the Turkish Government, and when war broke out she was on the point of leaving for Constantinople, when she was taken over.

The design of the Erin, Fig. 3, was settled by the three firms—Armstrongs, Vickers, and Brown—in consultation with the Turkish authorities, for whom the vessel was built, being commenced in November, 1911. In general characteristics she more nearly followed the "*King George V*" class than any other British ship, except that the secondary armament consisted of 6-inch guns, as in the "*Iron Duke*" class. This vessel also was taken over by the British Government in August, 1914, and certain modifications made to fit her for the British service. In respect of quantity of fuel carried, the *Erin* was below the standard adopted for vessels designed for the British Navy.

The third ship taken over from a foreign Government was ordered and commenced in 1911 at Elswick, from designs prepared by Mr. Perrett for the Chilean Government. There were two ships of the class, the *Almirante Latorre* (now the *Canada*), and the sister ship the *Almirante Cochrane* (now the *Eagle*). The *Canada*, Fig. 4, has ten 14-inch guns, twin-mounted, in the center line, and was originally designed to have twenty-two 4.7-inch as the secondary battery, but this was subsequently altered to sixteen 6-inch guns. The protection, again, was somewhat lighter than that of our Dreadnaughts, but the speed was rather higher, viz, 22¾ knots, and as a matter of fact this speed was considerably exceeded on trial. The ship was taken over by British Admiralty in September, 1914, and completed, after certain necessary modifications, a year later. Her fuel consisted of coal, with the addition of a certain amount of oil, as in most of our battleships. The sister ship, *Almirante Cochrane*, remained in an uncompleted condition on the stocks at Elswick till the spring of 1917, when she was taken over by the British Government and rearranged as an aircraft-carrying ship. She was renamed H. M. S. *Eagle*, and as a compliment to the United States Navy, she was, at the request of the Admiralty, launched by Mrs. Page, the wife of the late American Ambassador. The above record finishes the list of battleships proper which are completed for service during the war.

LARGE BATTLE-CRUISERS.

Coming to the battle-cruisers, particulars have already been published of all these down to H. M. S. *Tiger*. This ship was included in the 1911-12 program and followed on the *Queen Mary*, the general features of the

two ships being much alike, the chief differences being that the secondary armament of the *Tiger* is twelve 6-inch guns in lieu of sixteen 4-inch in *Queen Mary*, and *Tiger* has two submerged torpedo rooms, whereas the *Queen Mary* had only one.

After the design was approved by the Board, the ship was ordered and laid down at Clydebank on June 12, 1912, and completed in October, 1914. In common with so many of our ships completed during the war, the early commissioning and joining of the Fleet was so imperative, that no exhaustive trials in deep water were carried out, but the runs made on the Polperro Course showed that the designed power of 108,000 shaft horsepower could be obtained with little difficulty, corresponding to a speed of 30 knots. During the progress of the design, the oil-fuel capacity was very largely increased in case of need, the original tanks which only allowed for 1,100 tons were supplemented to admit of a maximum oil stowage of 3,480 tons, in addition to the 3,320 tons of coal; but it is not usual for the vessel to carry this full fuel stowage, at any rate, of oil.

BATTLE-CRUISERS "RENOWN" AND "REPULSE."

At the commencement of the war two additional battleships of slightly modified Royal Sovereign type, viz, the *Renown* and *Repulse*, had been laid down, but in view of the long time it would take to complete these ships, the construction was not pressed forward. Immediately after the battle of the Falkland Islands, in which our battle-cruisers *Invincible* and *Inflexible*, in company with other smaller cruisers, annihilated Von Spee's fleet, the value of the battle-cruiser type became very apparent, and on the initiative of Lord Fisher, then First Sea Lord, it was decided to stop the construction of the *Renown* and *Repulse* as battleships and to alter the design completely into that of very fast battle-cruisers. I received instructions to re-design these ships about Christmas, 1914. The new design had to give a speed of 32 knots, with the largest number of guns possible for such a vessel, and with protection similar to the *Invincible* and *Indefatigable* classes. A modified form of bulge was adopted in these ships to give additional underwater protection against torpedo attack. Proposals are now under way for still further adding to this bulge protection.

The general outline design was completed and approved in ten days, and six 15-inch guns were adopted as the main armament, the secondary armament consisting of seventeen 4-inch guns, of which fifteen were mounted in five specially-designed triple gun mountings. Owing to the circumstances, referred to above, it was necessary that the ships should be completed at the earliest possible date, and I suggested that the *Tiger's* machinery should be repeated with some additional boilers, and with the extra length it was found possible to obtain the speed of 32 knots, as laid down by the Board. Lord Fisher also insisted that the ships must be completed with fifteen months—an abnormally short time for an entirely new design, without any drawings prepared. This period of completion was not realized, although not greatly exceeded.

By January 21, 1915, the two firms entrusted with the orders, viz, John Brown and Co. and the Fairfield Company, were supplied with sufficient information to enable them to proceed with the structure, and both keels were laid on January 25. All the drawings and specifications were completed by April and the design finally approved in that month. The fuel was to be entirely oil, and with the additional boilers the power expected to be from 110,000 shaft horsepower to 120,000 shaft horsepower—the latter having actually been obtained on trial. The arrangement of the whole ship, showing the protection, is given in Fig. 5, the plating over the magazines being considerably increased during the construc-

TABLE I.—PARTICULARS OF BRITISH WARSHIPS CONSTRUCTED DURING THE WAR.

	Battleships.						Battle Cruisers.		Large Light Cruisers.	
	Iron Duke.	Queen Elizabeth.	Royal Sovereign.	Agincourt.	Erin.	Canada.	Tiger.	Renown.	Courageous.	Furious.
Length between perpendiculars	580 ft. 0 in.	600 ft. 0 in.	560 ft. 0 in.	622 ft. 0 in.	535 ft. 0 in.	625 ft. 0 in.	660 ft. 0 in.	750 ft. 0 in.	735 ft. 0 in.	750 ft. 0 in.
Length overall	622 ft. 9 in.	643 ft. 9 in.	624 ft. 3 in.	671 ft. 6 in.	550 ft. 6 in.	661 ft. 0 in.	704 ft. 0 in.	794 ft. 0 in.	786 ft. 3 in.	786 ft. 3 in.
Breadth, extreme	90 ft. 0 in.	80 ft. 6 in.	88 ft. 6 in.	89 ft. 0 in.	81 ft. 7½ in.	88 ft. 0 in.	90 ft. 6 in.	90 ft. 0 in.	81 ft. 0 in.	88 ft. 0 in.
Load draught, mean	28 ft. 0 in.	28 ft. 9 in.	28 ft. 6 in.	27 ft. 0 in.	28 ft. 6 in.	28 ft. 6 in.	28 ft. 6 in.	26 ft. 6 in.	23 ft. 8 in.	21 ft. 6 in.
Displacement in tons	25,000	27,500	25,750	27,500	22,000	22,000	28,500	26,500	18,800	19,100
Shaft horse-power of engines ..	29,000	75,000	40,000	24,000	26,500	37,000	108,000	112,000	90,000	90,000
Speed at load draught (knots) ..	21	25	23	23	21	22½	30	32 nearly	32	31½
Fuel at load draught (tons) ..	900	650	900	1,500	900	1,150	900	1,000	750	750
Coal capacity (tons)	3,250	—	—	5,200	3,120	3,200	3,220	—	—	—
Oil fuel capacity (tons) ..	1,050	2,400	2,400	620	710	550	3,450	4,250	2,250	2,400
Armament	10 13·5-in. 12 6-in. 4 21-in. T.T.	8 15-in. 12 6-in. 4 21-in. T.T.	8 15-in. 14 6-in. 4 21-in. T.T.	14 12-in. 20 6-in. 2 21-in. T.T.	10 13·5-in. 16 6-in. 4 21-in. T.T.	10 14-in. 14 6-in. 4 21-in. T.T.	8 13·5-in. 12 6-in. 4 21-in. T.T.	6 15-in. 17 4-in. 2 21-in. T.T.	4 15-in. 16 4-in. 14 21-in. T.T.	— 10 5·5-in. 18 21-in. T.T.
Armour—										
Side, amidships	12in., 9in., 8in.	13 in., 6 in.	12 in., 6 in.	9 in., 6 in.	12in., 9in., 8in.	9in., 7in., 4½in.	9 in., 6 in., 5 in.	6 in., 1½ in.	8 in.	8 in.
Side, forward and aft ..	6 in., 4 in.	6 in., 4 in.	6 in., 4 in.	6 in., 4 in.	6 in., 4 in.	6 in., 4 in.	4 in.	4 in., 3 in.	3 in. forward	3 in. forward
Bulkheads, forward and aft ..	9 in., 6 in., 4 in.	6 in., 4 in.	6 in., 4 in.	6 in., 3 in.	9 in., 6 in., 4 in.	4½ in., 4 in.	4 in., 2 in.	4 in., 3 in.	3 in., 2 in.	3 in., 2 in.
Barbettes	10 in. to 3 in.	10 in., to 4 in.	10 in. to 4 in.	9 in. to 3 in.	10 in. to 3 in.	10 in. to 4 in.	9 in. to 3 in.	7 in. to 4 in.	7 in. to 3 in.	7 in. to 3 in.
Gunhouses	11 in.	11 in.	11 in.	12 in., 3 in.	11 in.	10 in.	9 in.	11 in. to 7 in.	9 in., 7 in.	7 in.
Conning tower	11 in.	11 in.	11 in., 6 in.	12 in.	13 in.	11 in.	10 in.	10 in.	10 in.	10 in.
Protective—										
Vertical plating	1½ in., 1 in.	2 in., 1 in.	1½ in., 1 in.	1½ in., 1 in.	1½ in.	2 in., 1½ in.	2½ in., 1½ in. 1 in.	1½ in.	1½ in., 1 in.	1 in.
Protective deck	1 in. amidships 1 in.	1 in.	1 in.	1½ in. amidships 1 in.	1½ in. amidships 1 in.	1 in. amidships 1 in.	1½ in., 1 in.	1½ in., 1½ in.	1 in.	1 in.
Upper deck	2 in. to 1½ in.	2 in. to 1½ in.	1½ in. to 1½ in.	1½ in.	1½ in.	1½ in.	1½ in., 1 in.	½ in.	1 in.	1 in.
Main deck	1½ in. at ends	1½ in. at ends	2 in., 1½ in., 1 in.	1½ in., 1 in. at ends	1 in.	1½ in. aft	1 in. at ends	3 in. to 1 in.	1½ in., ½ in.	1½ in., ½ in.
Middle deck	2½ in., 1 in.	1 in. amidships	2 in. slope main to middle	1½ in., 1 in.	2 in., 1 in.	1 in.	—	—	—	—
Lower deck	2½ in., 1 in.	2 in., 1 in.	1 in., 2½ in., 2 in. 4 in.	2½ in., 1 in.	—	4 in., 2 in.	3 in., 1 in.	2½ in.	2 in., 1 in.	2 in., 1 in.

	Light Cruisers.				Monitors.						
	Aurora.	Calliope.	"D" Class.	Raleigh.	Humber.	14-in. Gun Class.	12-in. Gun Class.	Marshal Soult.	Erebus.	9·2-in. Gun Class.	6-in. Gun Class.
Length between perpendiculars	410 ft. 0 in.	420 ft. 0 in.	445 ft. 0 in.	505 ft. 0 in.	261 ft. 6 in.	320 ft. 0 in.	320 ft. 0 in.	340 ft. 0 in.	320 ft. 0 in.	170 ft. 0 in.	170 ft. 0 in.
Length overall	436 ft. 0 in.	446 ft. 0 in.	471 ft. 0 in.	605 ft. 0 in.	280 ft. 9 in.	334 ft. 6 in.	335 ft. 6 in.	405 ft. 0 in.	335 ft. 8 in.	177 ft. 0 in.	177 ft. 0 in.
Breadth, extreme	39 ft. 0 in.	41 ft. 6 in.	43 ft. 0 in.	65 ft. 0 in.	40 ft. 0 in.	90 ft. 0 in.	87 ft. 0 in.	90 ft. 3 in.	88 ft 0 in.	31 ft. 0 in.	31 ft. 0 in.
Load draught, mean	15 ft. 6 in.	13 ft. 6 in.	14 ft. 3 in.	17 ft. 3 in.	4 ft. 9 in.	10 ft. 0 in.	10 ft. 0 in.	10 ft. 3 in.	11 ft. 0 in.	6 ft. 0 in.	4 ft. 0 in.
Displacement in tons	3,500	3,750	4,650	9,750	1,200	6,150	5,600	6,670	6,000	540	535
Shaft horse-power of engines ..	40,000	40,000	40,000	40,000	1,450	2,000	2,500 to 2,500	1,500	6,000	480 to 960	480
Speed at load draught (knots) ..	29	29	29	29	12	6 to 7	6 to 7	6 to 7	12	12 to 12	12
Fuel at load draught (tons) ..	300	300	300	1,000	60	200	200	100	250	—	—
Coal capacity (tons)	—	—	—	2,000	197	400	340	—	—	—	—
Oil fuel capacity (tons) ..	310	300	1,000	1,000	60	—	—	295	750	30	45
Armament	2 6-in. 7 4-in. (1 H.A.) 2 21-in. T.T.	2 6-in. 6 4-in. 4 21-in. T.T.	6 6-in. 8 4-in. 4 21-in. T.T.	7 7·5-in. 12 3-in. H.A. (2 H.A.) 2 21-in. T.T.	2 6-in. 24·7-in. Mor.	2 14-in. 1 6-in. —	2 15-in. 1 to 4 6-in. —	2 15-in. 2 4-in. —	2 15-in. 2 4-in. —	1 9·2-in. 1 3-in. 2 —	2 6-pr. —
Armour—											
Side, amidships	4 in.	4 in.	4 in.	6 in.	4 in.	14 in.	14 in.	14 in.	14 in.	9 in.	9 in.
Side, forward and aft ..	4 in.	4 in.	4 in.	6 in.	4 in.	14 in.	14 in.	14 in.	14 in.	9 in.	9 in.
Bulkheads, forward and aft ..	4 in.	4 in.	4 in.	6 in.	4 in.	14 in.	14 in.	14 in.	14 in.	9 in.	9 in.
Barbettes	4 in.	4 in.	4 in.	6 in.	4 in.	14 in.	14 in.	14 in.	14 in.	9 in.	9 in.
Gunhouses	4 in.	4 in.	4 in.	6 in.	4 in.	14 in.	14 in.	14 in.	14 in.	9 in.	9 in.
Conning tower	4 in.	4 in.	4 in.	6 in.	4 in.	14 in.	14 in.	14 in.	14 in.	9 in.	9 in.
Protective—											
Vertical plating	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.
Protective deck	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.
Upper deck	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.
Main deck	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.	1 in.
Middle deck	—	—	—	—	—	—	—	—	—	—	—
Lower deck	—	—	—	—	—	—	—	—	—	—	—

	Submarines.						
	"H" Class.	"G" Class.	"E" Class.	"J" Class.	"K" Class.	"L" Class.	
Length between perpendiculars	150 ft. 0 in.	156 ft. 6 in.	164 ft. 6 in.	270 ft. 0 in.	324 ft. 0 in.	323 ft. 0 in.	
Length overall	161 ft. 0 in.	167 ft. 0 in.	171 ft. 0 in.	275 ft. 0 in.	339 ft. 0 in.	331 ft. 0 in.	
Breadth, extreme	22 ft. 6 in.	22 ft. 6 in.	15 ft. 9 in.	22 ft. 0 in.	26 ft. 6 in.	23 ft. 6 in.	
Load draught, mean	12 ft. 6 in.	13 ft. 3 in.	11 ft. 2 in.	14 ft. 0 in.	16 ft. 0 in.	13 ft. 6 in.	
Displacement in tons, surface	600	700	660	1,510	1,650	890	
Displacement in tons, submerged	800	970	800	1,880	2,650	1,070	
Horse-power of engines, surface	1,600	2,000	480	3,600	10,000	2,600	
Horse-power of engines, submerged	840	840	280	1,600	1,400	1,600	
Speed at load draught (knots) surface	15	14	10	19	24	15	
Speed at load draught (knots) submerged	10	10	10½	9	9	10	
Coal capacity (tons)	65	44	16	91	200	70	
Oil fuel capacity (tons)	1 24-in.	1 24-in.	—	1 24-in. or 4 4-in.	1 24-in., 1 24-in. H.A.	1 24-in. or 4 4-in.	
Armament	6 18-in. T.T.	6 18-in., 1 21-in. T.T.	4 21-in. T.T.	6 18-in. T.T.	6 18-in. T.T.	6 18-in. T.T.	

	Torpedo-Boat Destroyers.			T.B.D. Flotilla Leaders.		Patrol Boats.	Sloops and Mine-Sweepers.			"Chine" Gunboats.	
	"M" Class.	"B" and "S" Classes.	"V" and "W" Classes.	"Kempenfelt" Class.	"Scott" Class and "Shakespeare" Class (approximately same)	"P" Class.	Single-screw Sloops, "Flower" Class.	Paddle Mine-sweepers.	Twin-screw Mine-sweepers.	"Fly" Class.	"Insect" Class.
Length between perpendiculars	265 ft. 0 in.	265 ft. 0 in.	269 ft. 0 in.	325 ft. 0 in.	330 ft. 0 in.	230 ft. 0 in.	266 ft. 3 in.	225 ft. 0 in.	220 ft. 0 in.	180 ft. 0 in.	200 ft. 0 in.
Length overall	273 ft. 4 in.	276 ft. 0 in.	313 ft. 0 in.	335 ft. 0 in.	338 ft. 4 in.	244 ft. 0 in.	267 ft. 9 in.	245 ft. 0 in.	231 ft. 0 in.	196 ft. 0 in.	207 ft. 0 in.
Breadth, extreme	26 ft. 6 in.	26 ft. 0 in.	29 ft. 6 in.	31 ft. 0 in.	31 ft. 0 in.	23 ft. 0 in.	36 ft. 6 in.	29 ft. 0 in. (56 ft. over paddles)	28 ft. 0 in.	20 ft. 0 in.	20 ft. 0 in.
Load draught, mean	8 ft. 3 in.	8 ft. 0 in.	8 ft. 0 in.	10 ft. 0 in.	10 ft. 0 in.	7 ft. 7 in.	11 ft. 0 in.	6 ft. 9 in.	7 ft. 0 in.	2 ft. 6 in.	4 ft. 0 in.
Displacement in tons	1,555	1,555	1,800	1,612	1,800	672	1,250	810	760	82	245
Horse-power of engines	25,000	27,000	27,000	25,000	25,000 to 27,000	4,000	2,600	1,600	1,600	176	2,000
Speed at load draught (knots)	34	34	34	33	33	28	17	15	16	10	14
Coal capacity (tons)	140	150	195	200	200	80	150	—	—	—	25
Oil fuel capacity (tons)	—	—	—	—	—	—	300	160	140	5	—
	300	300	370	310	300	92	—	—	—	10	54
Armament	3 4-in.	3 4-in.	4 4-in. or 4 3-7-in.	4 4-in.	6 4-7-in.	1 4-in.	2 4-in. or 4 3-7-in.	1 3-in.	1 3-in.	1 4-in.	2 3-in.
	1 3-pdr.	1 3-pdr.	1 3-in.	2 3-pdrs.	1 3-in. H.A.	1 3-pdr.	2 3-pdrs.	1 6-pdr., 2 3-pdrs.	1 6-pdr., 2 3-pdrs.	1 3-in., 1 6-pdr.	2 3-in.
	4 21-in. T.T.	4 21-in. T.T.	4 or 5 21-in. T.T.	4 21-in. T.T.	6 21-in. T.T.	2 14-in. T.T.	—	—	—	—	—

Fig.1. H.M.S. "ROYAL SOVEREIGN"

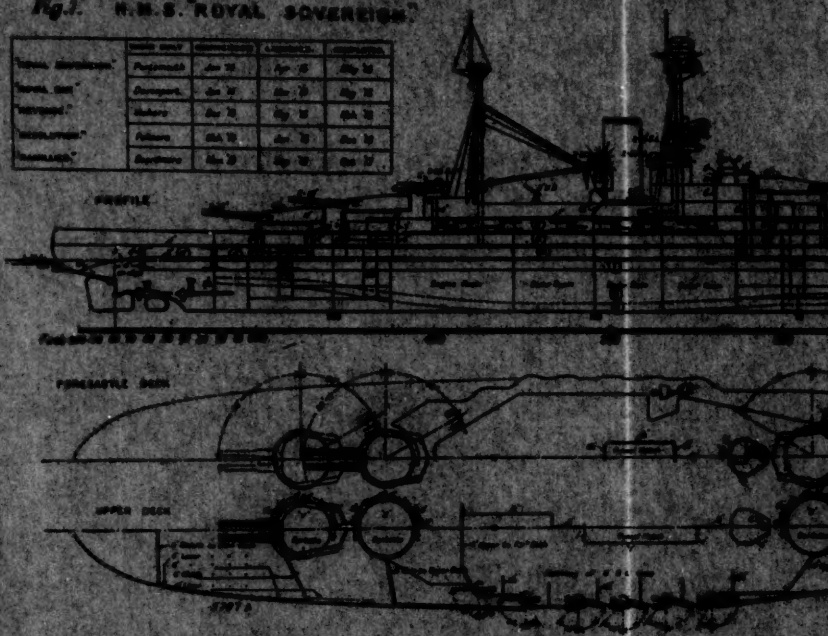
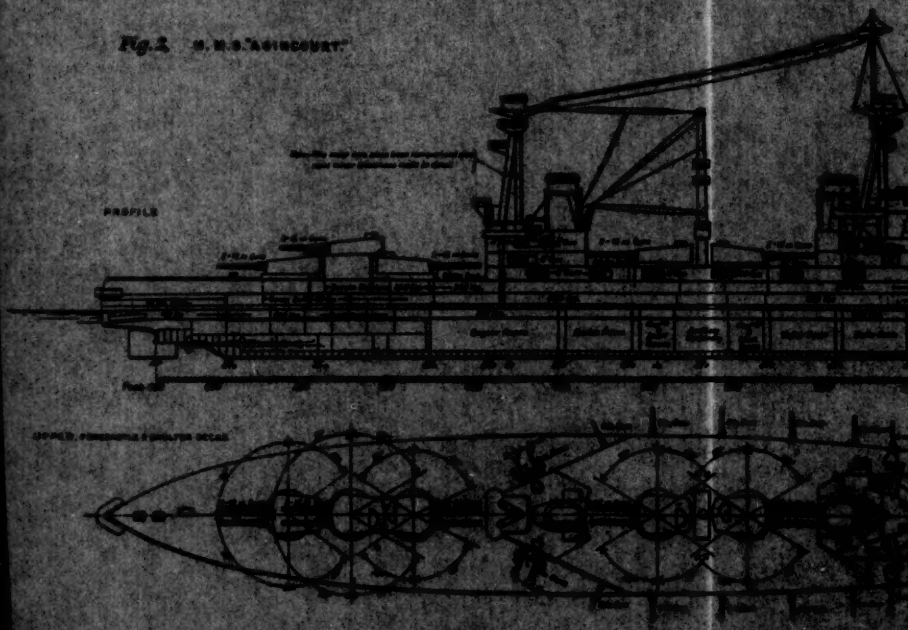


Fig.2. H.M.S. "AGINCOURT"



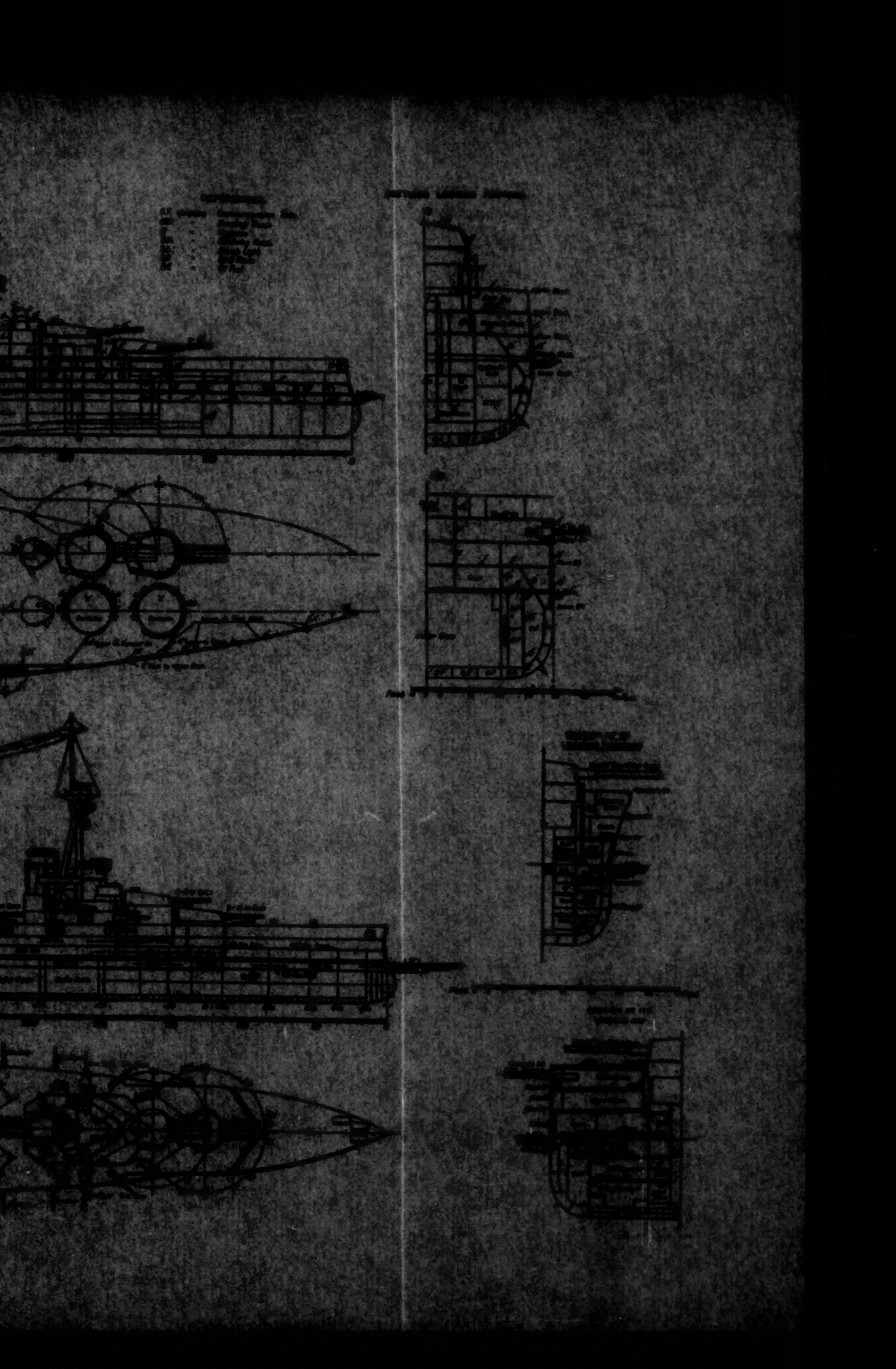


Fig. 3. H.M.S. "ERIN."

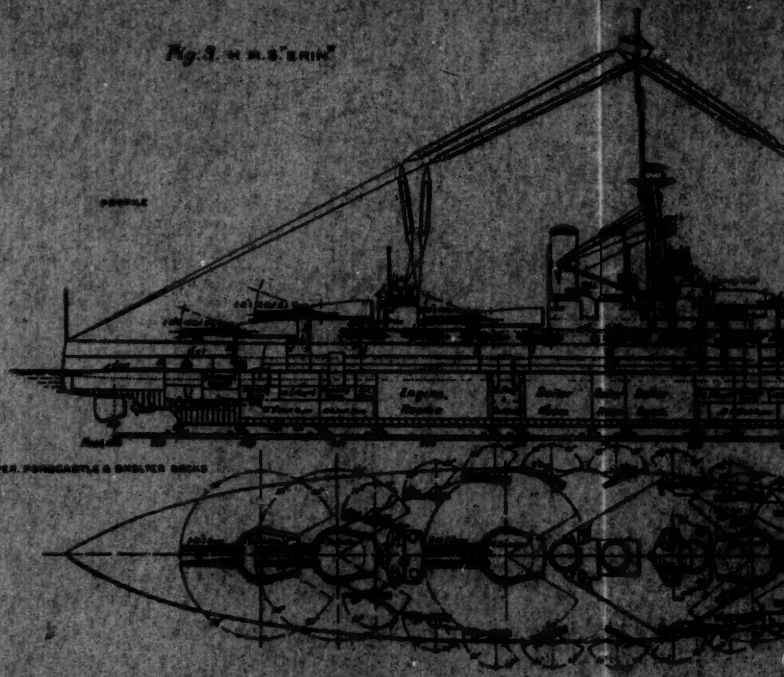
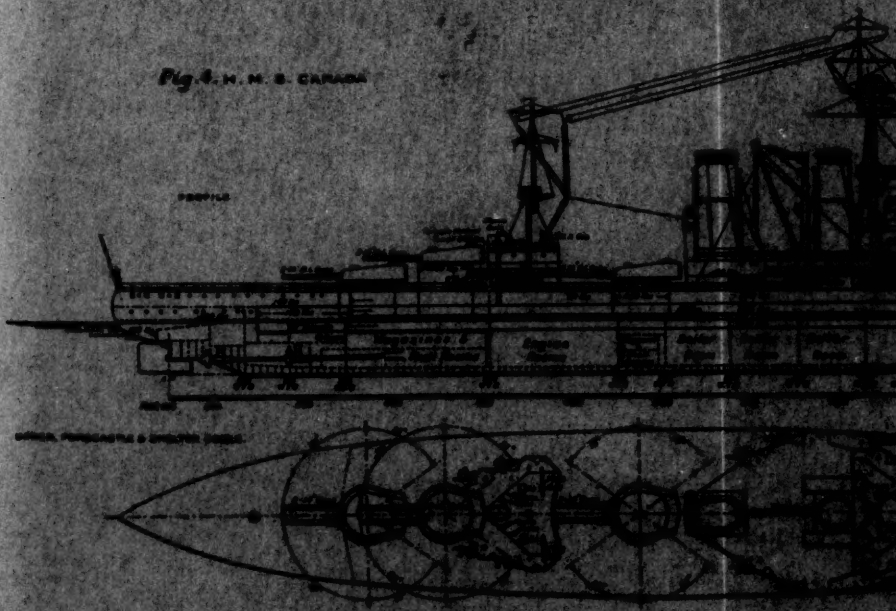


Fig. 4. H. M. S. CARADOC



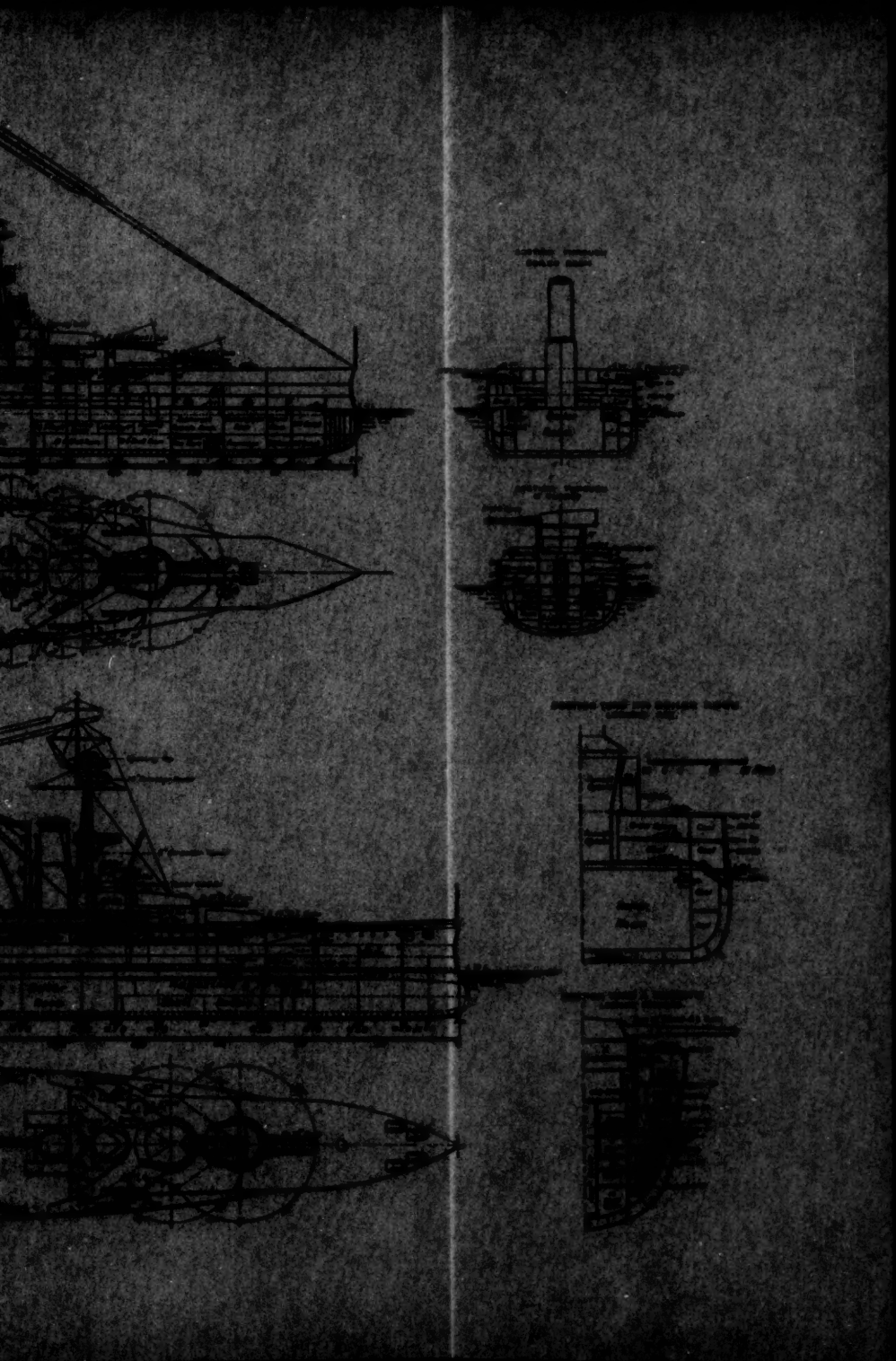


Fig. 4. BATTLE CRUISERS "REPULSE"

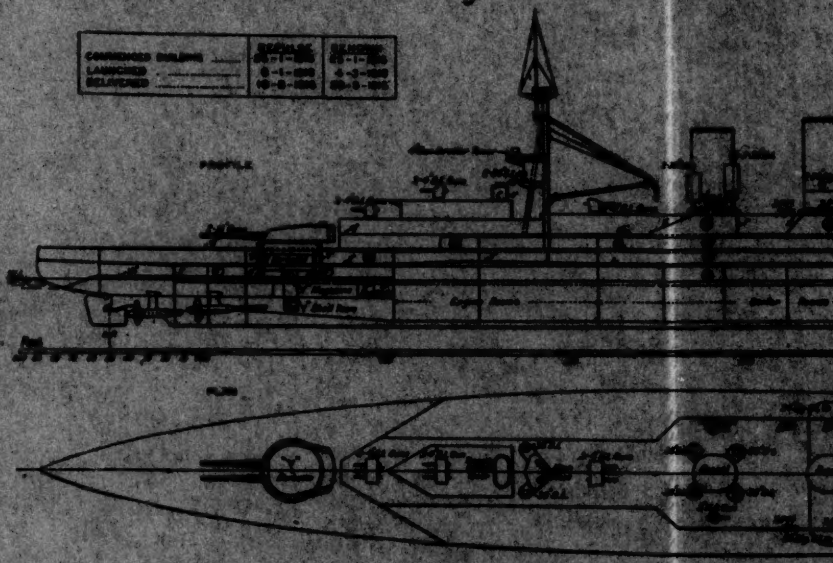
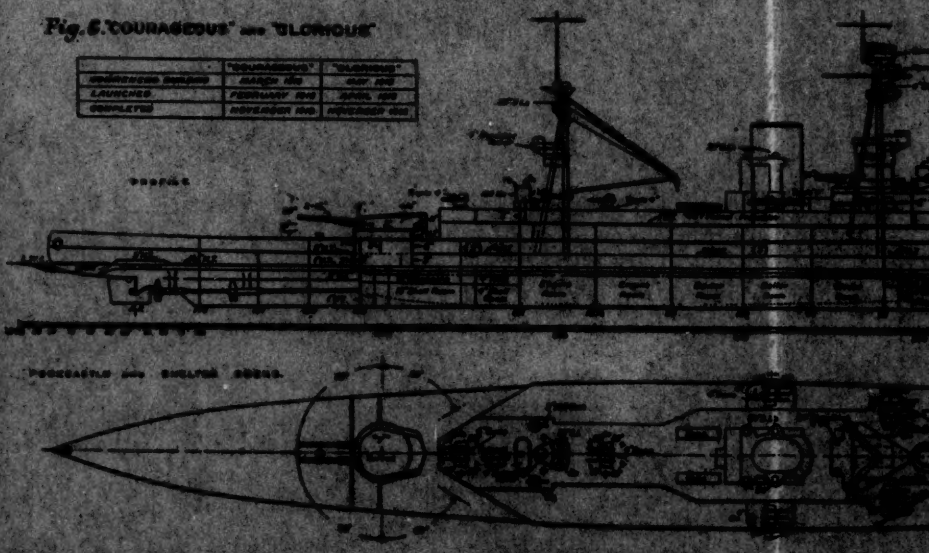


Fig. 5. "COURAGEOUS" and "GLORIOUS"



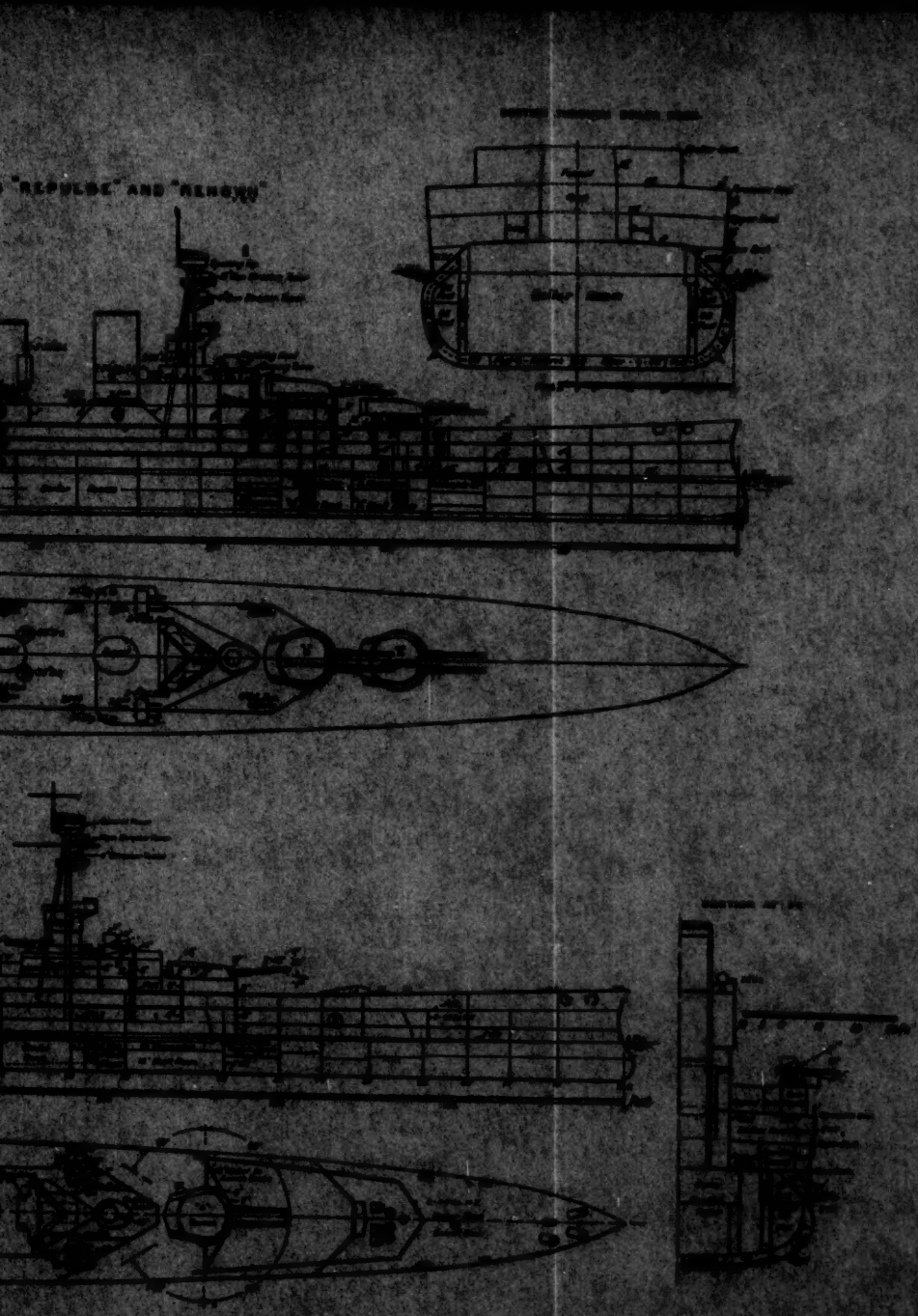


Fig. 7. FURIOUS

Commenced building	June 1915
Launched	August 1916
Completed	July 1917

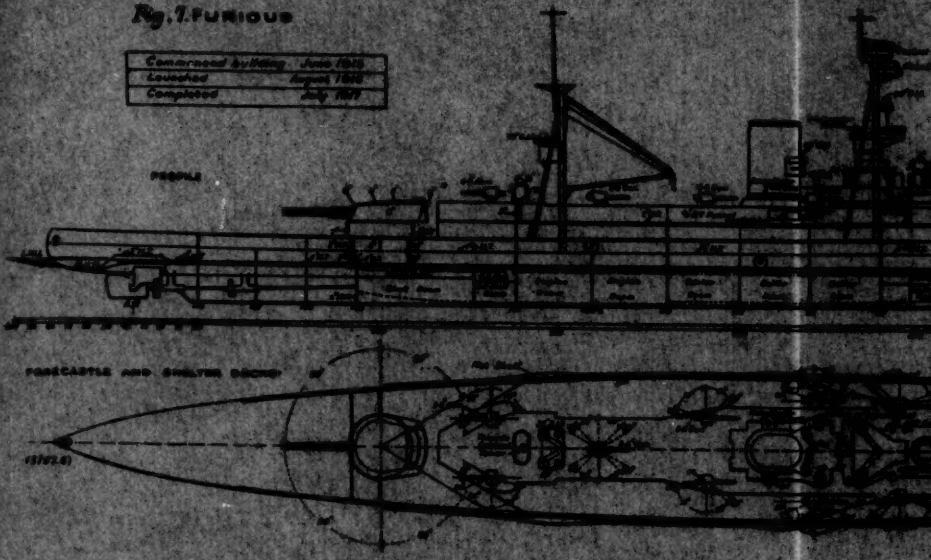
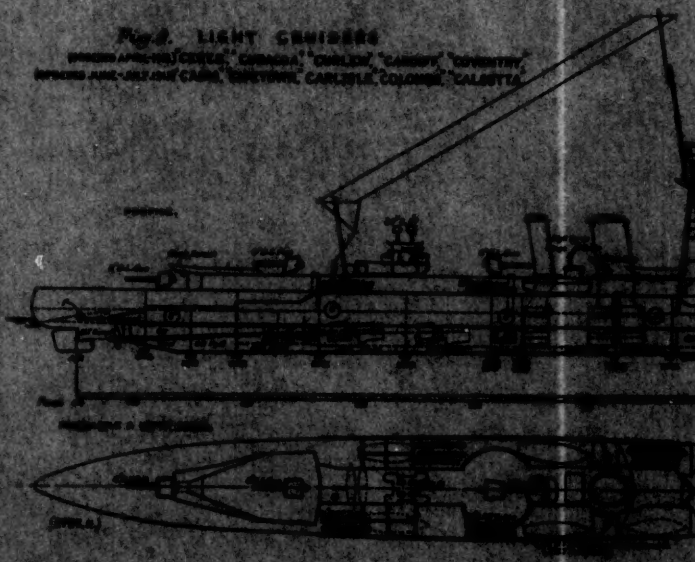
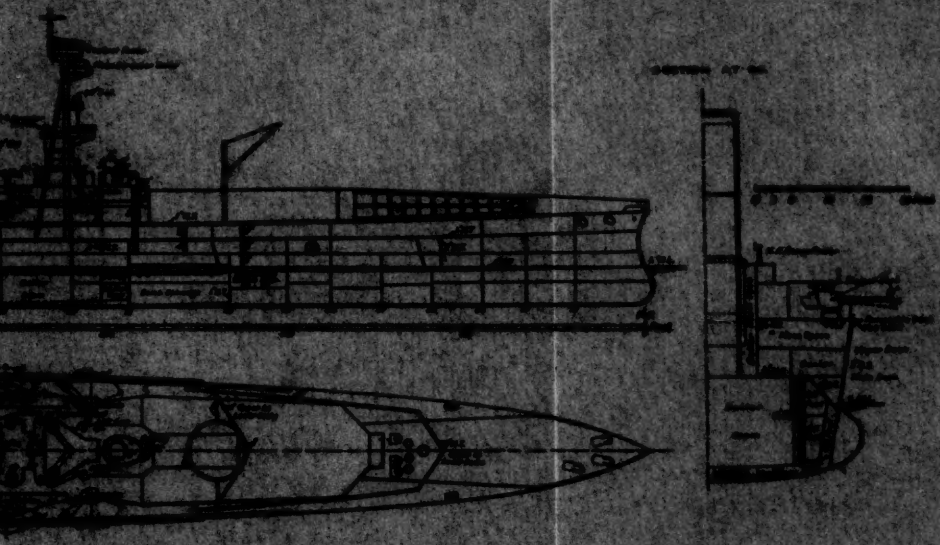


Fig. 8. LIGHT CRUISERS





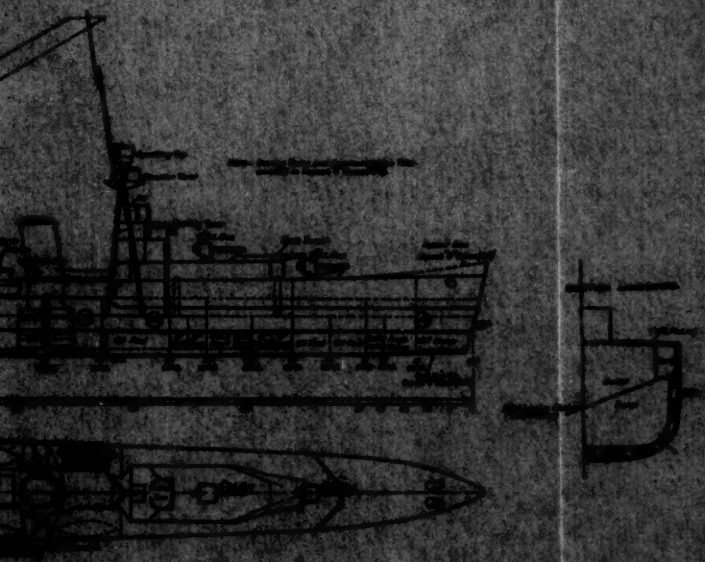


Fig. 9 LIGHT CRUISERS

ORDERED DECEMBER 1913

"RALEIGH" "LYNGHAW"

"HAWKINS" "FROBISHER"

"CAVENDISH"

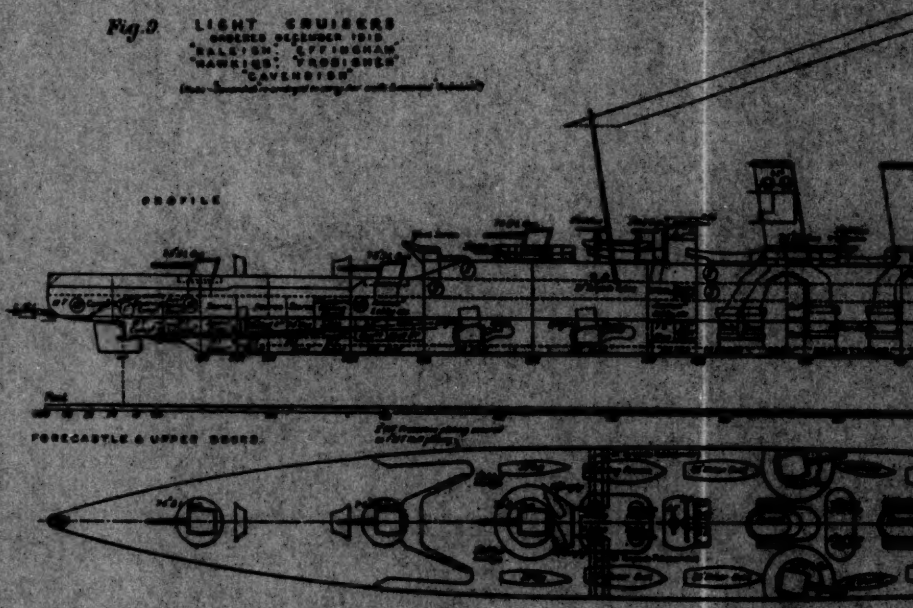
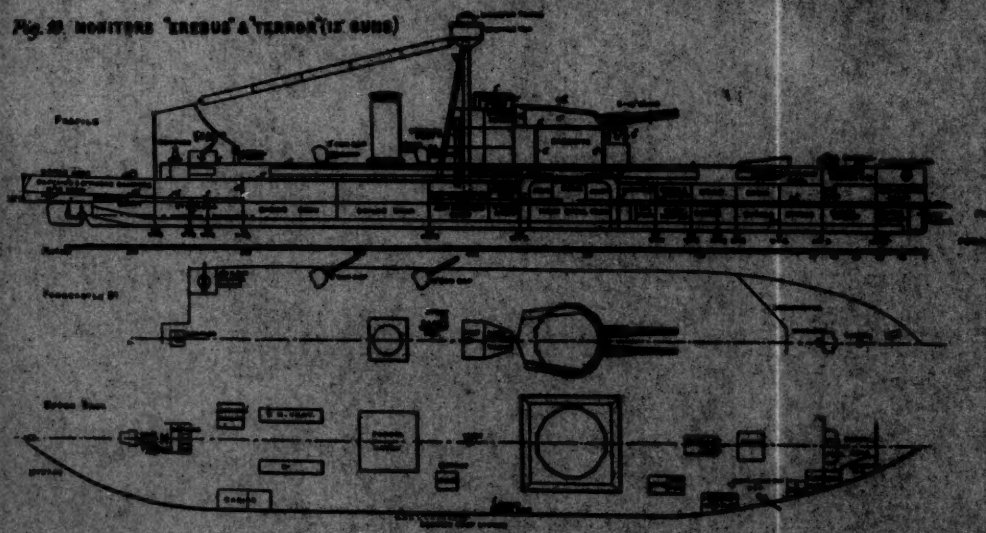
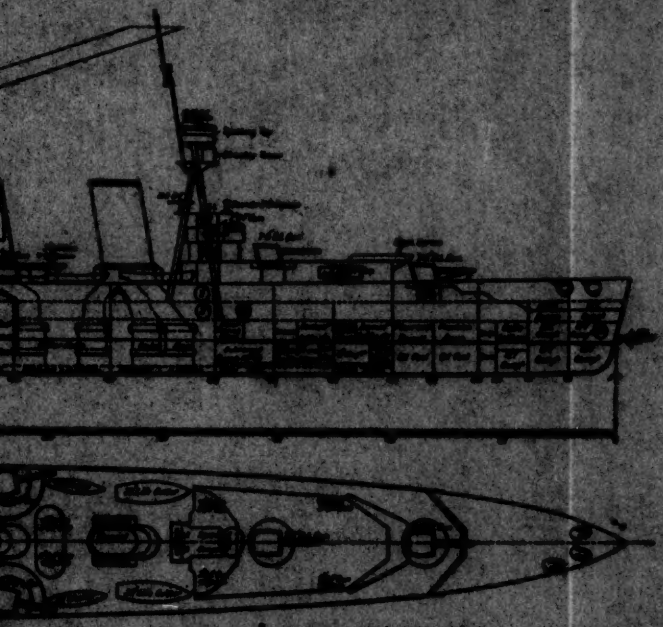


Fig. 10 MONITORS "EREBUS" & "TERROR" (15" GUNS)





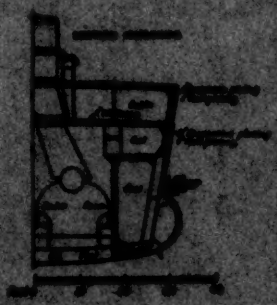
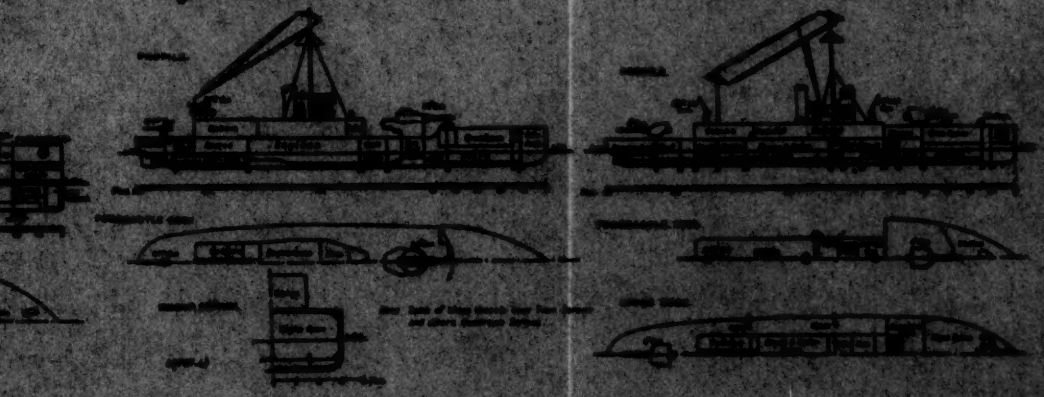


Fig.11. Э-Е MONITORS(N 15,20)

Ё MONITORS(N 25,30)



tion as a result of the Jutland fight. The *Repulse* was launched in January, 1916, less than a year from the laying down, and the *Renown* was launched two months later. The *Repulse* went through her commissioning trials early in August, and the *Renown* followed one month later and was completed in September. The speed of *Repulse* on trial was over 31¼ knots in the deep condition, and the *Renown* obtained 32.6 knots mean speed on the new measured course off Arran in the normal condition. The ships have been well reported upon at sea and maintain their speed well. I think that the construction of these vessels in a little over one and a-half years from the first order to get out the design, constitutes a record in design and construction of two such important vessels, and reflects great credit, not only upon the Royal Corps of Naval Constructors, but also upon the contractors and all concerned in the construction and completion of the vessels. In fact, the Admiralty conveyed their appreciation of this to me in a letter dated September, 1916.

LARGE LIGHT CRUISERS "COURAGEOUS," "GLORIOUS" AND "FURIOUS" (FIG. 6 AND FIG. 7).

Whilst the designs of the *Renown* and *Repulse* were in progress, I received instructions to design some very high-speed ships carrying powerful guns and of a size sufficient to keep their speed in moderate weather, but to have a draught lighter than any existing British or enemy ship of the same class, so as to be able to navigate shallow waters, if required.

As sanction was not likely to be obtained for building more capital ships taking two years or longer to complete, while additional light cruisers had been already approved of, it was decided to build the *Courageous* and *Glorious* on the lines of very light cruisers mounting a few guns of heaviest caliber, so as to be able to annihilate any enemy light cruisers or raiders. They were to have thin protection, similar to our light cruisers, and a speed of not less than 32 knots, the draught being restricted to about 22 feet, or about 5 feet less than any existing battleship or battle-cruiser carrying such heavy guns, the main armament of four 15-inch guns in two turrets, one forward and one aft, making them a match for any raider or light cruiser that might be encountered. At this time it should also be remembered that the armaments of ships, especially as regards heavy guns, had to be regulated by the guns and gun mountings which would be available or could be manufactured in the time at our disposal, and this condition applied to the 15-inch mountings which were adopted for these ships. The secondary armament consisted of eighteen 4-inch guns in six triple mountings, similar to the triple mountings of the *Renown* and *Repulse*. The side armor consisted of 2-inch protected plating on top of the 1-inch shell plating, as in our light cruisers, and a thin protective deck was worked all fore and aft, but this was considerably thickened over the magazines after Jutland. A modified bulge was arranged for, as in the *Renown* and *Repulse*.

The machinery adopted for these ships was of the type fitted in the light cruiser *Champion*. It consisted of a four-shaft arrangement of geared turbines, the power being transmitted to the propeller shafts by double helical gearing. The 18 boilers of Yarrow small-tube type were also similar to those of the light cruisers, and with all-oil firing a power of 90,000 shaft horsepower at about 340 revolutions was aimed at. Such trials as it was possible to make showed that 32 knots could easily be obtained at the designed displacement, and reports show that on service this is actually exceeded. The design of these vessels was begun late in January, 1915, and the order for one ship (*Courageous*) was placed with Messrs. Armstrong and the other (*Glorious*) with Harland and Wolff, the latter making their own machinery and Messrs. Parsons sup-

plying the machinery for Messrs. Armstrong's ship. It was intended that these vessels should be built in a year, or as near that as possible, but this was not realized, and the ships were both commissioned in October, 1916.

On her commissioning trials, the *Courageous* worked up to full power, and while steaming during the trials at full speed she met very heavy weather. Some signs of weakness were shown at the fore side of the forward turret where there is an inevitable discontinuity of longitudinal strength, and some doubling plates were accordingly added to the *Courageous*. Her sister vessel *Glorious* was in commission for over a year before similar additions were made to her, although no signs of weakness were shown. This incident shows that the very high speed obtained on trial, reaching 32 knots, should hardly be maintained against head seas in heavy weather.

The *Furious* was similar to, but a modification of, the *Courageous* and *Glorious*, having about the same length and the same machinery, but the form of midship section was somewhat different, having a more pronounced bulge and a simpler form of main framing and structure of the hull. The armament also was different, each turret, instead of having two 15-inch guns, was arranged to carry one big gun of 18-inch bore, although arrangements were made to substitute pairs of 15-inch guns, if thought desirable later.

The order for this ship was placed with Messrs. Armstrong about two months after that of *Courageous*, and she was to be finished in the shortest possible time. Early in the spring of 1917, however, the necessity for having fast aeroplane-carriers became very obvious, and it was approved to fit *Furious* for this purpose. This entailed doing away altogether with the fore turret and making other considerable alterations. A large hangar was built on the forecastle deck, and a flying-off platform 160 feet long was arranged on the roof of the hangar, which was designed to house about ten machines. Later it was decided to remove the after turret as well, and a flying-on deck 300 feet long, extending from the funnel aft, was constructed. The secondary armament, which had consisted originally of eleven 5½-inch guns, was retained, with the exception of one gun; the remaining 10 guns being rearranged. Four sets of triple 21-inch torpedo tubes were fitted on the upper deck aft, and one pair each side on the upper deck forward. After these alterations were completed the ship was tried and commissioned in July, 1917, a speed of 31½ knots being obtained with 94,000 shaft horsepower at 330 revolutions.

LIGHT CRUISERS (FIG. 8 AND FIG. 9).

Following upon the previous light cruisers of the "*Town*" classes, the particulars of which have already been published, a very important departure was made in the light-cruiser design in the program 1912-13, when the "*Arethusa*" class was designed by Sir Philip Watts. The importance attached to speed was specially brought out in this design, and it was decided to instal very powerful machinery with a shaft horsepower of 40,000, and this could only be achieved by adopting a type of engines and boilers closely approximating to those hitherto used for destroyer classes.

In conjunction with high speed a good armament was provided, consisting of two 6-inch and six 4-inch guns, though in the original design the armament consisted entirely of 4 inches. The ship's sides up to the level of the upper deck were protected by specially high-tensile plating varying from 2 inches to 1½ inches throughout the machinery spaces, in addition to the 1-inch shell plating. This arrangement of plating also greatly added to the strength and stiffness of the ship regarded as a girder. Further particulars of the class are given in the Tables. The

Arethusa and other light cruisers were in the action off Heligoland on August 28, 1914, and gave an excellent account of themselves.

In the 1913-14 program the "*Calliope*" class, slightly larger vessels than the *Arethusa*, but with the same power, were decided upon. After considerable discussion regarding the merits of mixed or homogeneous armament, it was decided to give these vessels two 6-inch guns, both on the center line placed aft, and eight 4-inch guns. The protection consisted, as in the previous design, of a 2-inch belt over the shell plating, making a total thickness of approximately 3 inches. Most of this class had practically the same machinery as the *Arethusa*, but Parsons geared turbines were installed in two of them, the *Calliope* having four shafts and the *Champion* having only two shafts. This was at the time a very important experiment, the putting of 20,000 H.P. through gearing being a very bold departure from anything which had been hitherto contemplated. The final results obtained with *Champion* were, however, excellent, and she obtained a speed of 29½ with 337 revolutions and about 41,000 shaft horsepower, this speed being slightly in excess of any of the other vessels of the class at corresponding displacement.

For the subsequent classes I would refer to the Tables, which show a gradual growth in size and power of armament; "*Ceres*" class, Fig. 8, finally having a length of 425 feet and a beam of 43 feet 6 inches, and a normal displacement of about 4,200 tons. These vessels carried five 6-inch guns, all on the center line.

The next class were the "*D's*" the general arrangement and protection of which followed that of the "*Ceres*," except that six 6-inch guns were carried on the center line instead of five. The power was only slightly increased in these ships over the previous classes, but the revolutions were reduced to 275, all these later classes having the twin-screw geared arrangement, and although the displacement of the "*D's*" increased to 4,650 tons, the additional length and the reduction of revolutions enabled the speed of close upon 30 knots of the whole class of light cruisers, "*C's*" and "*D's*," to be practically maintained. The "*Arethusas*" and the "*C*" and "*D*" classes all had oil fuel only.

In addition to these light cruisers, which were all to Admiralty design, two vessels—the *Birkenhead* and *Chester*—which were built at Messrs. Cammell Laird's for the Greek Government, were purchased in 1915. These vessels were considerably heavier than the "*C*" class, and more closely resembled the British "*Chatham*" class. They carried an armament of ten 5½-inch guns. The machinery was modified to burn oil only, in the *Chester*, instead of coal and oil as in the *Birkenhead*, and the resulting increase in power to 31,000 gave the former a speed of 26½ knots.

The "Raleigh" Class.—In the summer of 1915 designs were prepared for a considerably heavier class of light cruisers, more especially designed for ocean work in any part of the world. They were to have a speed of 30 knots and a large radius of action. Various armaments were considered, and it was finally decided to adopt an armament of seven 7.5-inch guns with twelve 3-inch (four being on high-angle mountings). Five of the big guns were placed on the center line, as shown in Fig. 9, and the other two were on the broadsides amidships. The bow and stern guns were superposed, thus giving a fire of four guns, both ahead and astern, and six guns on either broadside. The conditions laid down involved a ship of great length, and these vessels consequently have an overall length of 605 feet. These ships were originally designed to burn oil and coal, but this was subsequently altered in three ships of the class to all oil, the original power of 60,000 shaft horsepower on a four-shaft geared turbine arrangement being considerably increased up to about 70,000 shaft horsepower by the change to all-oil firing. These vessels also

different from the light cruisers referred to above in having modified bulges as protection against underwater attack. The form of the section shows how these were arranged. The protective plating was similar to that of the other light cruisers.

MONITORS. (FIGS 10 AND 11.)

The first vessels of this type to be added (or reintroduced) to the British Navy were the three ex-Brazilian river monitors, built by Messrs. Vickers, Limited, and taken over by the British Government in August, 1914, and renamed *Humber*, *Mersey* and *Severn*. The particulars of these vessels are given in the Tables. It will be seen that the armament consisted of medium caliber guns, viz, 6-inch and 4.7-inch. These vessels, though lightly built, have done very good service in the war, both on the East African and Belgian coasts.

The need for vessels of the monitor type mounting heavy guns soon became apparent, and I received instructions in November, 1914, to prepare designs of monitors of more substantial structure for sea-going service, but of light draught with good protection and carrying some heavy guns, the light draught combining the advantages of being able to go close inshore and of greatly reducing risk of being struck by a torpedo.

The earliest design was that of the 14-inch gun monitors, four in number, which was commenced in 1914. Four twin-mounted 14-inch guns and mountings were available, and with the very simple form of structure adopted, these vessels were designed and built in six months. They were quickly followed by the 12-inch monitors, which were of similar design but carried pairs of 12-inch guns, generally taken from older battleships. These vessels were also built in about six months. They all had a complete bulge of a form which was of simple construction, with an air-space outboard and a waterspace between that and the ship proper. This bulge was carried well fore and aft, and the ships being of rather bluff form, the speed was somewhat below that estimated; the bluff form of the bulge having a bad effect upon the propeller performance. However, no particular point had been made about the speed, and we had rather to take what machinery we could get in the shortest possible time.

Following on the 12-inch monitors, early in June, 1915, two more vessels were ordered, mounting a pair of 15-inch guns. For these ships, internal-combustion engines, which were well under way, but designed for another purpose, were installed. In September, 1915, two improved 15-inch monitors were ordered and named the *Erebus* and *Terror* (Fig. 10). These were of finer form, of more power and greatly increased speed. These vessels also had an improved form of bulge, though it did not differ essentially from the bulges of the earlier monitors. The *Erebus* and *Terror* were designed to have a speed of 12 knots when fully laden, but they actually are capable of about 14 knots.

Following the earlier 15-inch monitors, some much smaller vessels, each carrying a 9.2-inch gun, were designed, and others again which carried 6-inch guns (Fig. 11). I need not refer at any length to these, as the general particulars are given in the table. A good many of both large and small monitors went out to the Dardanelles in the early part of the war and did very good work, and for a long time they seemed to bear a charmed life, as they enjoyed complete immunity from torpedo attack. Whether this was due to their shallow draught or to the enemy submarines thinking it was useless to torpedo the bulges, I do not know. Later, however, the *Erebus* and *Terror* were both torpedoed: the latter received three torpedoes forward, but got to port all right, though severely damaged, two of the torpedoes hitting forward of the bulge; the third, which hit the bulge itself, did very little damage. The former ship was hit full amidships by a distance-controlled boat carrying a very heavy charge, but

the bulge gave her complete protection and she was repaired in less than a fortnight. It is interesting to note in this connection also that some of our old cruisers of the "*Edgar*" class, which had had bulges added to them very early in the war, were torpedoed in the Mediterranean, but the bulge gave complete protection to the ship proper. They were taken to port and repaired. Of the heavier monitors it may be remarked that of all ships carrying heavy guns these vessels were probably more often in action off the Belgian coast and elsewhere than any of our heavy-gun ships, and they no doubt gave the enemy in occupation of that coast a very anxious time.

DESTROYERS AND FLOTILLA LEADERS.

To give a complete account of the development of the design and construction of our destroyers and flotilla leaders during the war would mean a very long paper, and I think, therefore, I must refer only to the tables and plans (Figs. 12, 13 and 14) which I give. These vessels have gradually increased in size and power, and war requirements have continually added to the weights which they have had to carry, including considerably more fuel, heavier armament, both of guns and torpedoes, depth charges, larger bridges, and other additions. In fact, some of the ships which before the war were 900-ton vessels, now carry an additional 100 tons, which, of course, is a very heavy handicap to these vessels when expected to maintain a high speed. The introduction, however, of the geared turbine has added enormously to the efficiency of the machinery and propellers; the lower revolutions which can be introduced with the gearing enabling us to adopt much more suitable propellers, in addition to giving a greater economy of fuel.

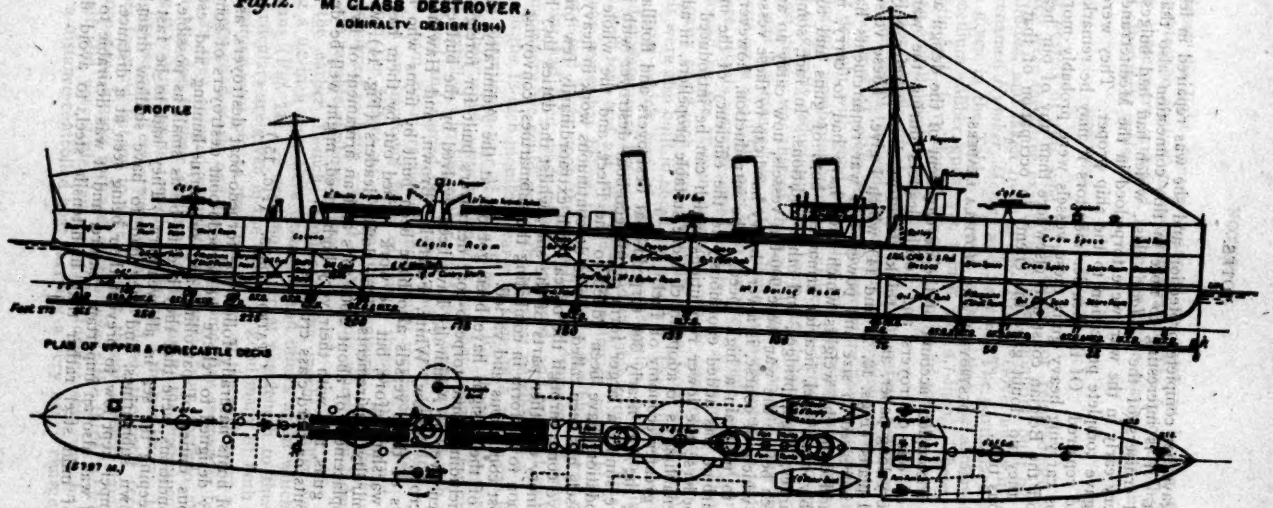
During the war nearly 300 torpedo-boat destroyers and flotilla leaders, which are simply a larger form of torpedo-boat destroyer with improved accommodation, have been added to the Fleet, and the whole class of these vessels has been called upon to do continuous work in heavy weather. They have come through the ordeal with extraordinarily few breakdowns of machinery or other parts of the ship, whilst the duties they have been called upon to perform in combating the submarines, convoying, &c., have been most continuous and varied.

Most of the designs have been got out at the Admiralty, but certain other, including some torpedo-boat destroyers built for foreign countries and taken over by Admiralty have been evolved by the builders, Messrs. Thornycroft, Yarrow, White, Fairfield, Brown, and Hawthorn Leslie. Numbers of these vessels also have been built by firms who had never built a warship before, but the work turned out by them has fully met the Admiralty requirements. The flotilla leaders (Fig. 14), with a deep load displacement of about 2,000 tons and an armament of five 4-inch or 4.7-inch guns, and with their very high speed, might well be described as fast scouts or third-class cruisers.

PATROL BOATS. (FIG. 15.)

Patrol boats naturally follow the torpedo-boat destroyers, as they were specially designed to relieve the torpedo-boat destroyers of some of their strenuous duties of patrolling and submarine-hunting and escort work. The conditions were that they had to be as small as possible, consistent with keeping the sea in all weathers. They had to be fast enough to run down submarines, and they had to have shallow draught and all top-hamper kept low to prevent their being seen at a distance. Economy of fuel was also an important feature, and it was desirable to have them built of mild steel rather than high-tensile steel, to avoid drilling and

Fig.12. "M" CLASS DESTROYER.
ADMIRALTY DESIGN (1914)



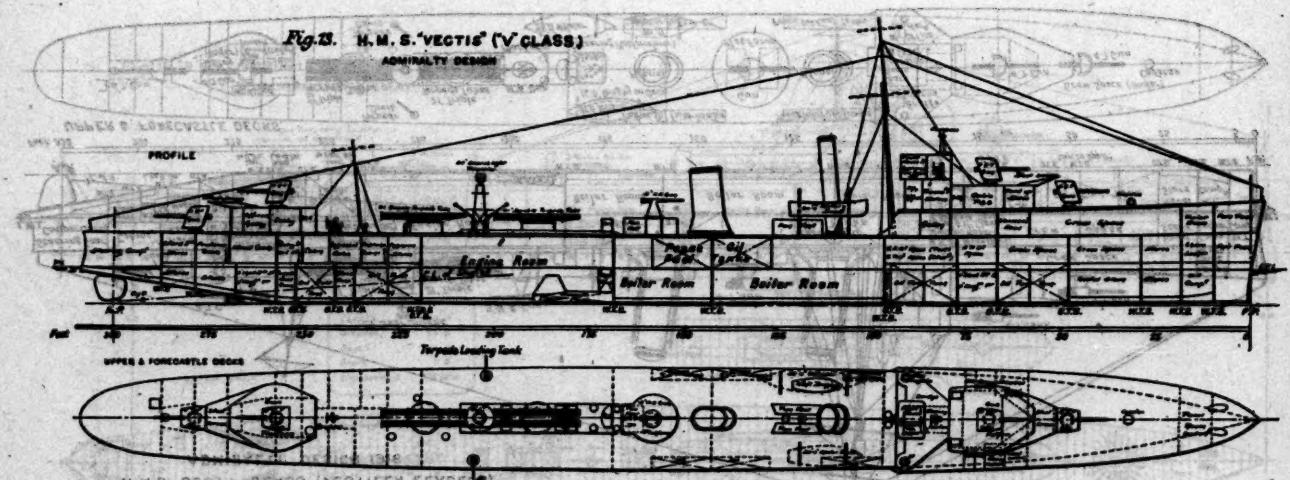
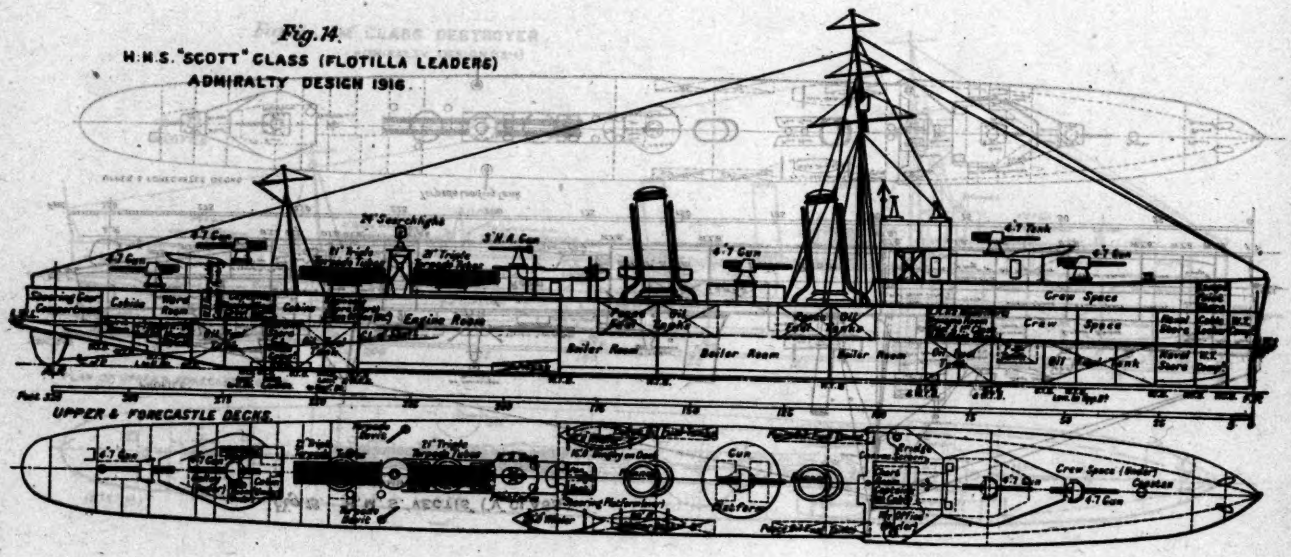


Fig.13. H. M. S. "VECTIS" ('V' CLASS)
ADMIRALTY DESIGN

Fig. 14.

H.M.S. "SCOTT" CLASS (FLOTILLA LEADERS)
ADMIRALTY DESIGN 1916.



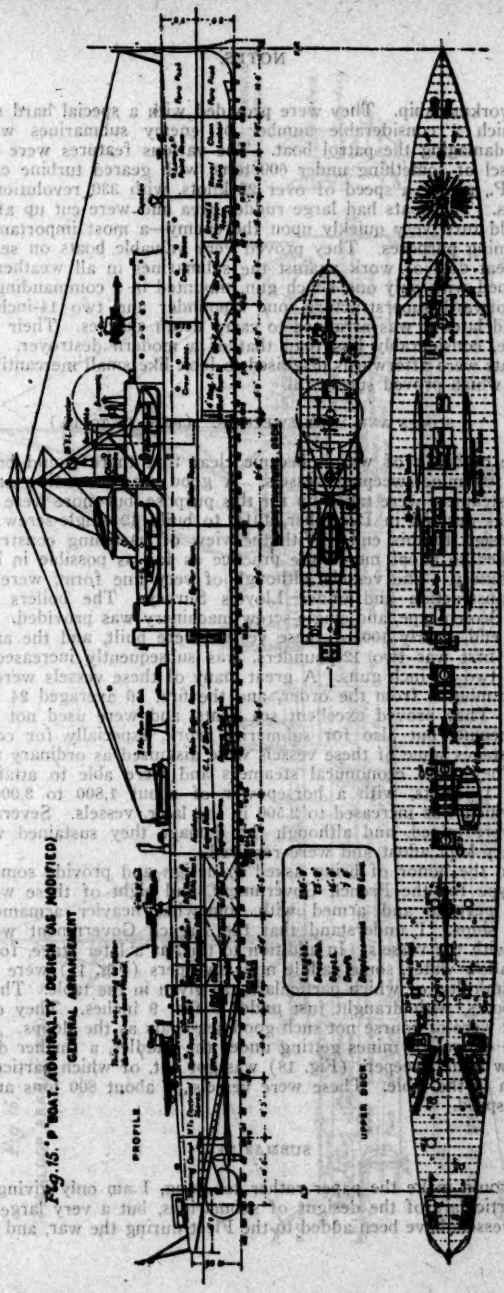


Fig. 15. 'P' BOAT. ADMIRALTY DESIGN (AS MODIFIED)
GENERAL ARRANGEMENT

special workmanship. They were provided with a special hard steel ram, with which a considerable number of enemy submarines were sunk without damaging the patrol boat. The various features were combined in a vessel of something under 600 tons, with geared turbine engines of 3,800 H.P., giving a speed of over 22 knots, with 330 revolutions of the propellers. The boats had large rudder area and were cut up aft so that they could turn very quickly upon the enemy—a most important feature for ramming purposes. They proved very valuable boats on service and did a great deal of work against the submarines in all weathers. They were armed with only one 4-inch gun, mounted in a commanding position in the forward superstructure, one 2-pounder and two 14-inch torpedo tubes, and later it was arranged to carry depth charges. Their cost was, of course, considerably less than that of a modern destroyer. Some of these boats were afterwards disguised to look like small mercantile craft—a device which proved successful.

SLOOPS AND MINE-SWEEPERS. (FIGS. 16 TO 18.)

On the outbreak of war it became clear that there would be a great demand for mine-sweeping vessels. A good many coasting and cross-Channel steamers were taken up for this purpose, but more were required, and it was decided in December, 1914, to build 12 single-screw ships of simple design to this end. With the view of hastening construction, it was decided to adopt mercantile practice as far as possible in both hull and machinery. The vessels, although of very fine form, were built of simple construction and under Lloyd's Survey. The boilers were of ordinary Scotch type, and single-screw machinery was provided.

In the end, nearly 100 of these vessels were built, and the armament, which at first was two 12-pounders, was subsequently increased to two 4-inch or two 4.7-inch guns. A great many of these vessels were built in about six months from the order, and the first 36 averaged 24 weeks in building. They proved excellent sea boats, and were used not only for mine sweeping, but also for submarine work, especially for convoying. At later stages some of these vessels were disguised as ordinary merchant ships. They were economical steamers and were able to attain a full speed of 17 knots, with a horsepower of about 1,800 to 2,000 in the earlier, which was increased to 2,500 in the later, vessels. Several of the vessels were mined, and although the damage they sustained was very severe, they kept afloat and were repaired.

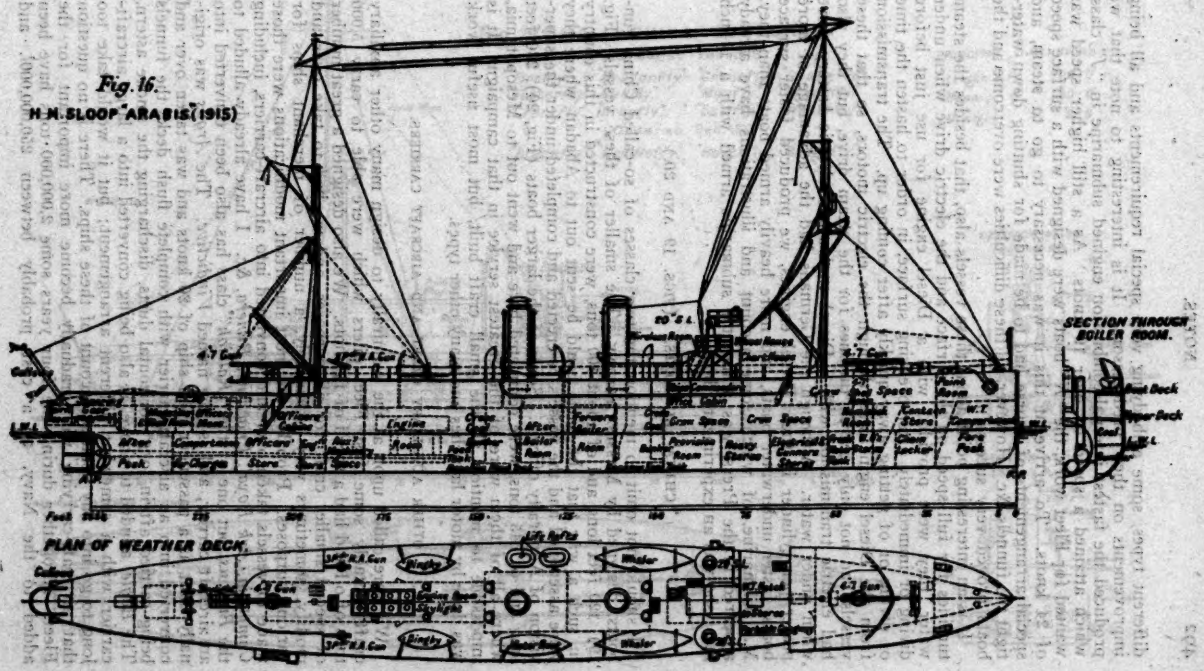
We had the honor of being asked to design and provide some vessels of this type for the French Government, and eight of these were built for that purpose and armed with somewhat heavier armament than our own ships. I understand that the French Government were very satisfied with the vessels. In addition to this, at a later stage, for sweeping in shallow water, some paddle mine sweepers (Fig. 17) were designed at the Admiralty, of which particulars are given in the table. These were 15-knot boats, with draught just under 6 feet 9 inches. They did good work, but were of course not such good sea-boats as the sloops. As there was some danger of mines getting under the paddles, a further design of twin-screw mine sweepers (Fig. 18) was got out, of which particulars are also given in the table. These were vessels of about 800 tons and about 16 knots speed.

SUBMARINES.

As it would make the paper rather too long, I am only giving a table of the particulars of the designs of submarines, but a very large number of these vessels have been added to the Fleet during the war, and some 12

Fig. 16.

H.M. SLOOP "ARABIS" (1915)



different types, some embodying very special requirements and all being improvements on their predecessors. It is interesting to note that we produced the fastest internal-combustion engined submarine in "*J*" class, which attained a speed of over 19 knots. As a still higher speed was wanted for Fleet work, the "*K*" boats were designed with a surface speed of 24 knots. To arrive at this it was necessary to go to steam, and special arrangements of course had to be made for shutting down water-tight the funnels, &c. However, all these difficulties were overcome and the boats proved very successful.

It is an interesting point about these vessels also, that besides the steam turbines for full speed on the surface and the electric drive when under water, they were provided with a Diesel engine for use just before diving or immediately after breaking surface, in order to hasten the time of diving or of getting away quickly after coming up. The transmission from the Diesel engine was through the electric motors, so that these vessels had not only geared turbines for the steam drive, but they also had electric transmission with a Diesel engine and electric battery drive when under water. Although the Germans had the advantage of more power per cylinder in their Diesel engines, we produced faster surface boats, faster underwater boats, and more heavily armed boats than they. M.1 submarine, of which some account and illustrations have already appeared in the Press, was a monitor submarine armed with a 12-inch gun; she was an experimental boat.

"CHINA" GUNBOATS. (FIGS. 19 AND 20.)

I should not omit to mention the two classes of so-called China gunboats, designed by Messrs. Yarrow. The smaller of these vessels (Fig. 19), 120 feet long and of about 100 tons, were constructed in this country in such a way that the parts could be sent out to Abadan where they were assembled and the vessels re-erected and completed under the supervision of Admiralty officers. Some of the larger boats (Fig. 20), 230 feet long and of 645 tons, were completed here and went out to Mesopotamia, where all of them were of the greatest service in that campaign. It is impossible to mention all the small craft built, but most useful work was done by motor launches and many other types.

OTHER AUXILIARY CRAFT AND AIRCRAFT CARRIERS.

We were called upon at the Admiralty to design many other auxiliary craft—notably some fast Fleet oilers which were able to carry 5,000 tons of oil and had a speed of 15 knots. We also designed a great number of special smaller craft for all purposes, and we had to take over and modify, sometimes very extensively, a number of merchant ships for various purposes. Probably the most important modifications were those made to vessels taken over and converted into aircraft-carriers, including *Campania, Ark Royal, Engadine, Riviera,* &c. I have already alluded to the *Furious,* but one of the "*Raleigh*" class has also been converted into an aircraft carrier, and is now named *Vindictive.* The *Argus* was originally built as a passenger mail ship of 20 knots and was taken over and converted into an aircraft-carrier with complete flush deck, the funnels being carried aft in long horizontal ducts, discharging the smoke astern. The *Eagle,* alluded to above, is also being converted into a large aircraft-carrier with a somewhat different arrangement; but it would take too long to go into any detailed account of these ships. There is no question that air-craft-carrying will gradually become more important for the Fleet. Altogether during the four years some 2,000,000 tons have been added to the Navy, at a cost probably between 250,000,000*l.* and

Fig. 19. SMALL CHINA GUNBOATS.

Ordered Feb. 1915.

"Firefly." "Dragonfly" "Greenfly" "Snakefly."
"Butterfly." "Cadfly." "Mayfly." "Stonefly."
"Cranefly." "Brayfly." "Sawfly." "Waterfly"

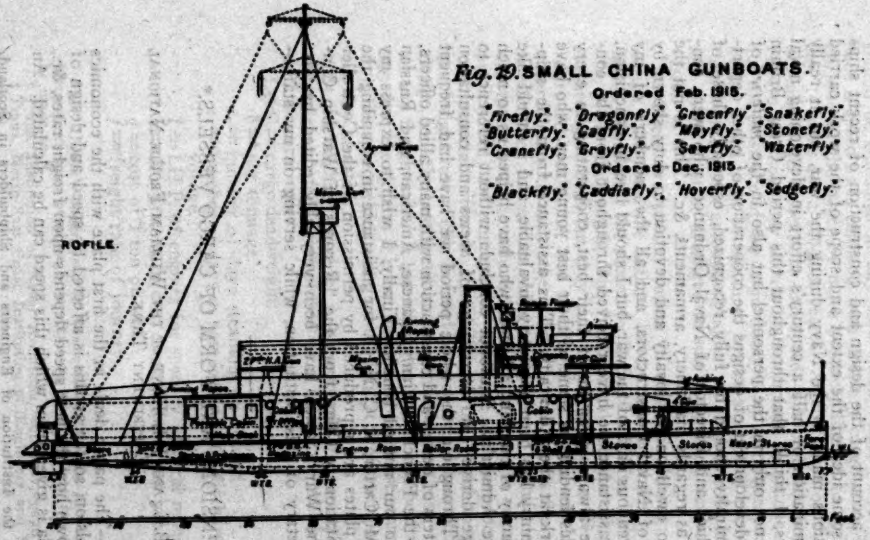
Ordered Dec. 1915

"Blackfly." "Caddisfly." "Hoverfly." "Sedgefly."

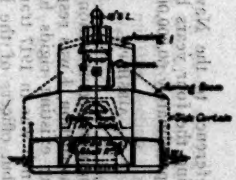
PRINCIPAL DIMENSIONS.

Length, B.P.	120'-0"
Beam	20'-0"
Draught	2'-6"

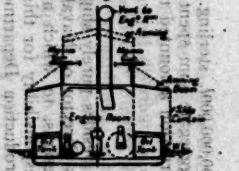
PROFILE.



SECTION AT 22. Looking Aft.



SECTION AT 36. Looking Forward.



MAIN & BATTERY DECKS.

300,000,000*l.* sterling. Reference to the Navy Estimates show that the aggregate sum spent during the four years before the war on new construction amounted to approximately 60,000,000*l.*

AIRSHIPS.

Besides purely ship design and construction, up till the autumn of 1917, as Director of Naval Construction, I was responsible for the rigid airship designs, to which all our existing rigids have been constructed. The responsibility for this branch was in 1917 transferred to a special Airship Production Department, but officers of the Royal Corps of Naval Constructors transferred to the new department continued the work they had been doing on airships.

The above brief account of the design and construction of recent ships will, I hope, give some idea of the extent and scope of the work carried out and the additions made to the Navy during the war, but it really represents the culmination of half a century's effort in developing naval design; and shows, I think, that throughout this period Great Britain had led the way, not only in the personnel but also in the *materiel* of the Navy. In the development of designs the coöperation of other departments of the Admiralty should be fully recognized, especially those of the Engineer-in-Chief and Director of Naval Ordnance, who were consulted throughout as regards machinery, armaments, &c. This is not the occasion for me to dwell on the loyalty and devotion to duty shown by the Royal Corps of Naval Constructors, and all the members of my staff during the strenuous days of the war, but I should like, in conclusion, to say how much assistance we have received throughout from the contractors, who have always given us of their best, coöperating in every possible way, besides lending us some of their best young men, who have done excellent work at the Admiralty. This assistance from the shipbuilders of the country has been absolutely invaluable, and I should like to take this opportunity of thanking all those who have helped us so much in our work at the Admiralty. I cannot conclude without reference to the great advantage derived by interchange of views and consultation with our Allies. Throughout the whole period we have had frequent conferences on matters of design and construction with many allied officers, including those of the French, Italian, Japanese, American, and Russian Navies, certainly to our mutual benefit. Finally, I wish to express my thanks to Mr. C. M. Carter, R. C. N. C., for assistance in compiling the tables given. The plates are reproduced by permission of the Controller of His Majesty's Stationery Office from the "Records of Warship Construction during the War," which have been very ably edited by Mr. R. W. Dana, Secretary of this Institution, while serving on my staff.—"Engineering."

SPEED, DIMENSIONS AND FORM OF CARGO VESSELS.*

By G. S. Baker, O.B.E., and J. L. Kent, of the William Froude National Experiment Tank.

§1. *Introductory.*—The paper deals in the first place with the economics of cargo-ship propulsion, so far as this is affected by speed and design of hull form. It is shown how the best speed depends upon freight rates, &c., and a simple formula is given from which this speed can be calculated. An

* Paper read before the Institution of Engineers and Shipbuilders in Scotland, February 18, 1919.

example in Appendix I gives best speeds varying from 12 knots to 14 knots for a 4,000-mile voyage. The second part of the paper (§10 to §19) gives in detail the various propulsive considerations which have to be borne in mind in settling the area of midship section, longitudinal distribution of displacement, and shape of level lines and body sections. Finally, by permission of the Controller-General of Merchant Shipbuilding, some notes are given on straight-frame ships, based upon test work carried out for His Majesty's Government. Both the latter parts of the paper are illustrated by giving typically good lines and results for curved and straight-frame ships.

§2. *Dimensions and Speed of Cargo Vessels.*—The primary purpose of this paper is to describe and define the necessary features which can or should be embodied in any slow-speed vessel. By slow speed is meant such a speed as is common with ordinary cargo tramps, and may for the purpose of the paper be more precisely limited to speeds not exceeding, say, 14 knots for a 400-foot vessel. As the shape of the vessel necessarily depends on its dimensions and speed, it is desirable first to consider these factors and how they should be determined in practice. The only reason for the existence of the cargo tramp is its earning capacity. The efficiency of the vessel must be measured by the amount of profit earned in a given time with a given amount of capital. For this to be high, the dimensions and speed must be arranged to suit the general service conditions of the vessel.

§3. There are several papers in existence dealing with this phase of design. Of these the authors have examined three:* the first written by Mr. J. Hamilton,† the second by E. Saxton White, B.Sc.,‡ and the third by Mr. John Anderson.§ The fundamental idea in all these papers is undoubtedly correct, but they tackled the problem under very different and restricted conditions, and so arrived at results which are not always in accordance with each other. In all these papers the total yearly expense is calculated for various ships or voyages of certain lengths, and the conditions necessary to reduce this to a minimum (in one case with a standard profit added) are enumerated. But none of these papers takes into account one important item in the ability of the vessel to earn profit, namely, the freight rate. This is fixed by economic factors quite apart from ship design, and the best speed and dimensions are those which make the most of any freight rate. It is a misnomer to call this best speed the economical speed of the ship, as it is not necessarily the most economical result in coal. It is the speed which will give the largest profit per pound capital invested, per day invested.

§4. In the first case, let it be assumed that an existing vessel of displacement Δ tons is employed on a service of which the average voyage is L nautical miles.

Let C = cargo deadweight.
δ = net tonnage.
n_2 = number of days steaming on voyage.
n_1 = number of days in port loading and discharging, repairs, &c., per voyage.
I = I.H.P. of engines for speed V nautical miles per day.
k = No. of tons of coal consumed per day per I.H.P.
q = cost of 1 ton of coal.
f = freight rate of journey.

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* Mr. Unwin's paper, read in Newcastle-on-Tyne on January 14, 1919, which is on better lines than previous ones, came to hand too late for any specific reference to be made to it.
† Trans. Institution of Naval Architects, vol. xxiv, page 258.
‡ Trans. North-East Coast Institution of Engineers and Shipbuilders, vol. xxviii, page 29.
§ Trans. Institution of Naval Architects, vol. ix, page 23.

x = tonnage dues in pound per ton for complete journey.
t = sum of brokerage, management, loading, discharge in pounds per ton of cargo, for complete journey.
P = price of vessel in pounds.
yP = sum of insurance, repair, depreciation, wages per year, in pounds.

Then the rate of earning money per day per pound of capital is:

$$M = \frac{(f-t)\,C - x\delta - q\,k\,I\,n_S}{(n_S + n_L)\,P} - \frac{y}{365} \quad . \quad . \quad (1)$$

Writing $(f-t)\,C - x\,\delta = r$

and $R = q\,k\,l V^3 \Delta^{\frac{2}{3}}$ = cost of coal burnt per day in pounds.

I may be written = $l V^3 \Delta^{\frac{2}{3}}$

and $n_S = \frac{L}{V}$,

Hence $$M = \frac{V r - q\,k\,l\,\Delta^{\frac{2}{3}}\,L\,V^3}{(L + n_L V)\,P} = \frac{y}{365} \quad . \quad . \quad . \quad . \quad (2)$$

For this to be a maximum, when the speed is changed, the first differential with regard to V must be zero.

Or $$R = \left(\frac{r}{3\,n_S + 2\,n_L}\right) \text{ or } = \frac{(f-t)\,C\,x\,\delta}{3\,n_S + 2\,n_L} \quad . \quad . \quad . \quad (3)$$

If this value of R be substituted in the expression for M, the maximum profit at the best speed in pounds per day per pound invested becomes:

$$M = \frac{2\,R}{P} - \frac{y}{365} \quad . \quad . \quad . \quad . \quad (4)$$

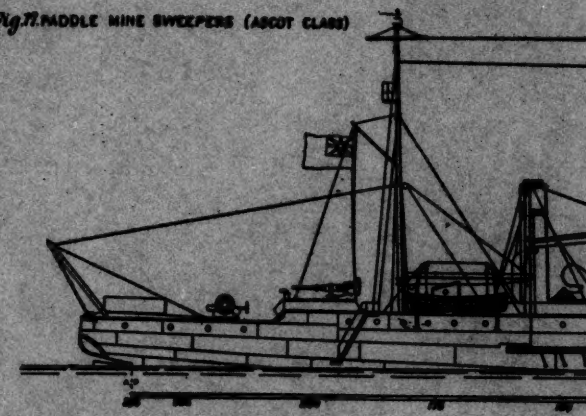
or, the profit per day earned by the ship = (twice the cost of coal burnt per day) − (capital charges per day)

§5. Equation (3) shows how the cost of coal burnt per day R should be varied according to the freight rate f, and how the coal burnt per day, and therefore the best speed, should rise and fall with length of journey. Equation (1) shows the importance of reducing n_1 the number of days in port to a minimum. These formulæ can be put into various forms, but the above are simplest for handling. In passing, it should be mentioned that Mr. Macfarlane Gray, in the discussion on Mr. Hamilton's paper, gave what amounted to equation (4) above, without the capital charges item.

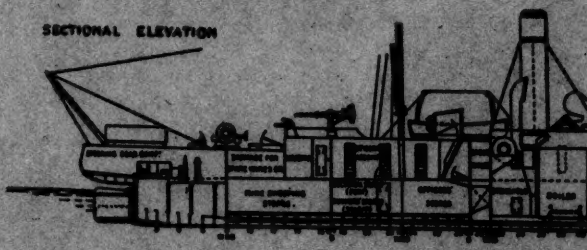
§6. The second case is the determination of best dimensions. Theoretically, the treatment may be the same as before, *i.e.*, the first differential of M with regard to displacement Δ must be zero. But since almost each term in equation (1) is dependent on displacement, the resulting equation is too cumbersome to be of use. The best procedure, therefore, is to take equation (3) and find the appropriate R and V for several displacements, and then to find which displacement gives the highest value of M in equation (4) for all reasonable freight rates and coal prices.

§7. To illustrate this portion of the paper, and to introduce and familiarize the designer with the use of the formula, the detail working of an example is given in Appendix I. Three widely-different freight rates and five widely-different vessels are chosen, and M, the rate of earning profit per pound capital per day, is worked out for all of them. It will be seen that:

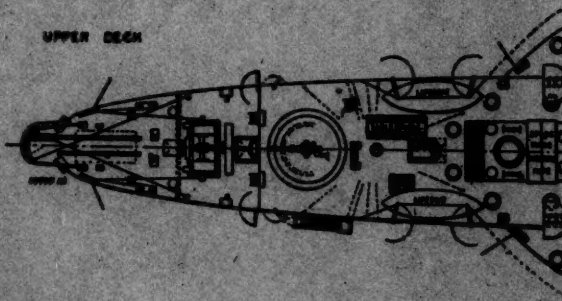
Fig.77. PADDLE MINE SWEEPERS (ASCOT CLASS)



SECTIONAL ELEVATION



UPPER DECK



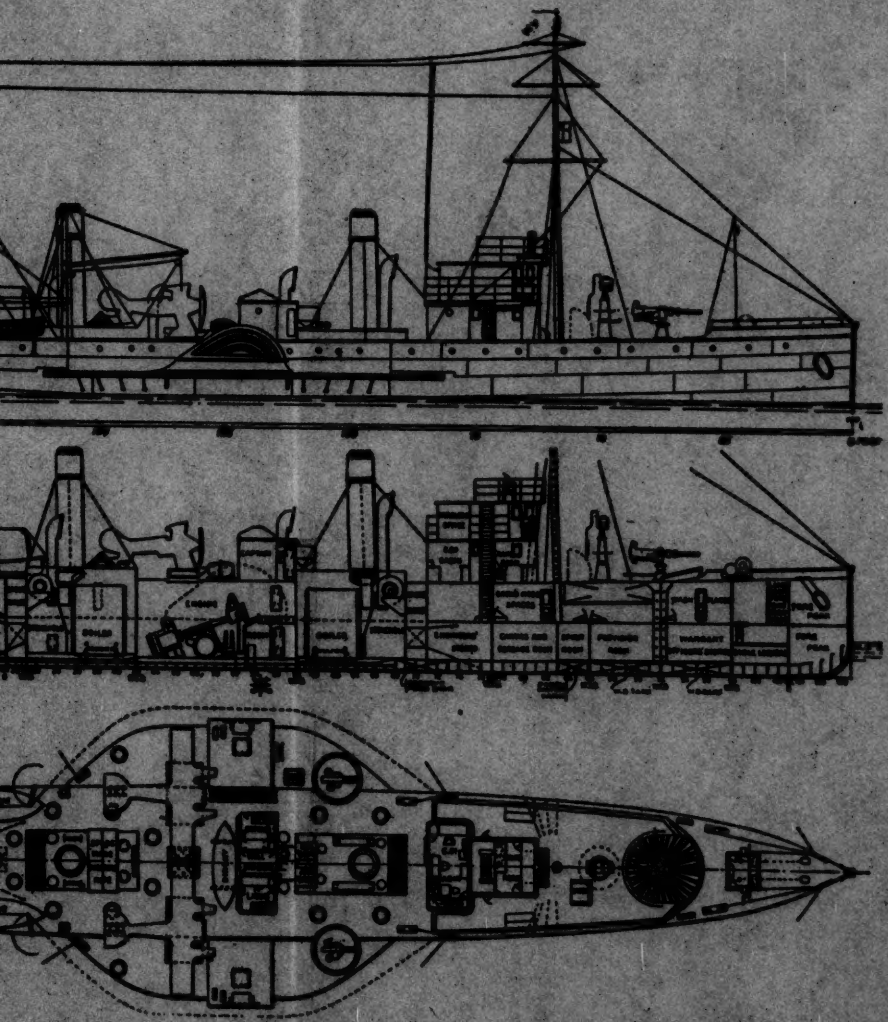
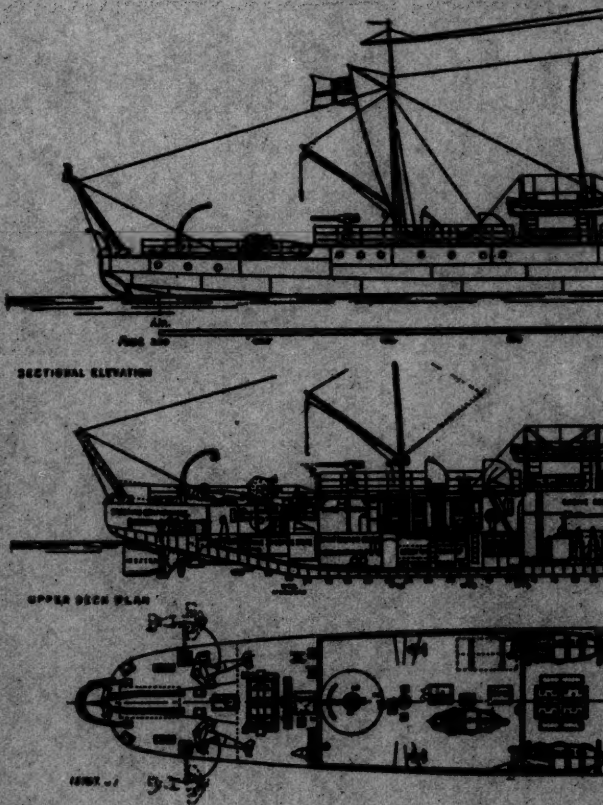


Fig. 18. TWIN SCREW MINE SWEEPERS



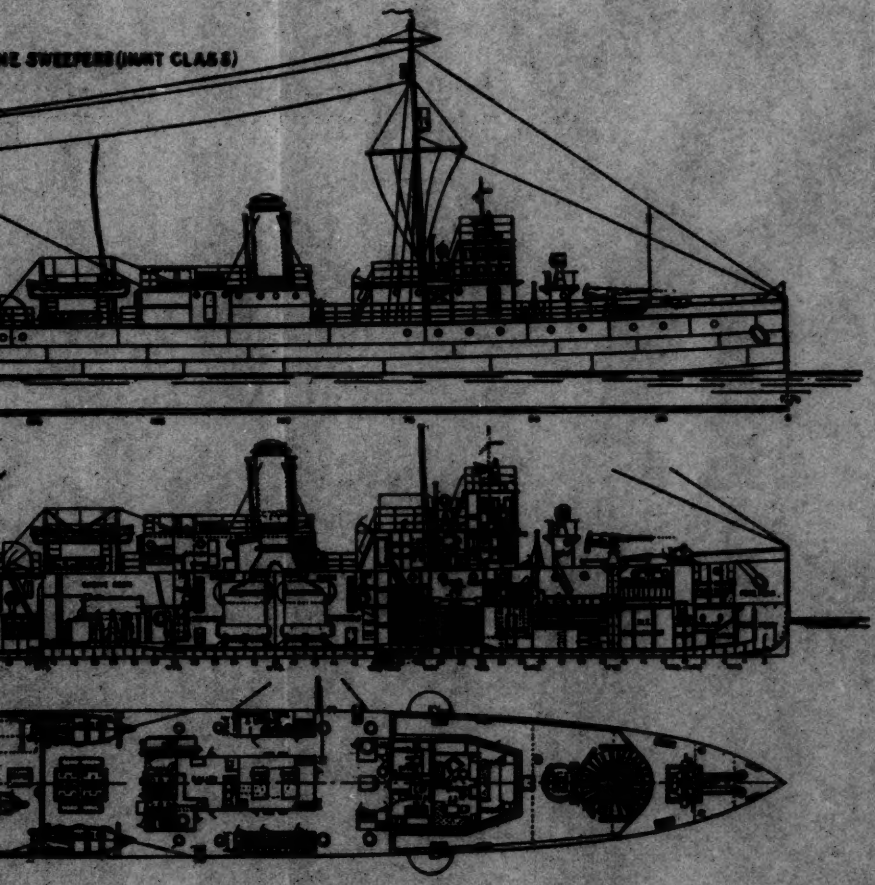
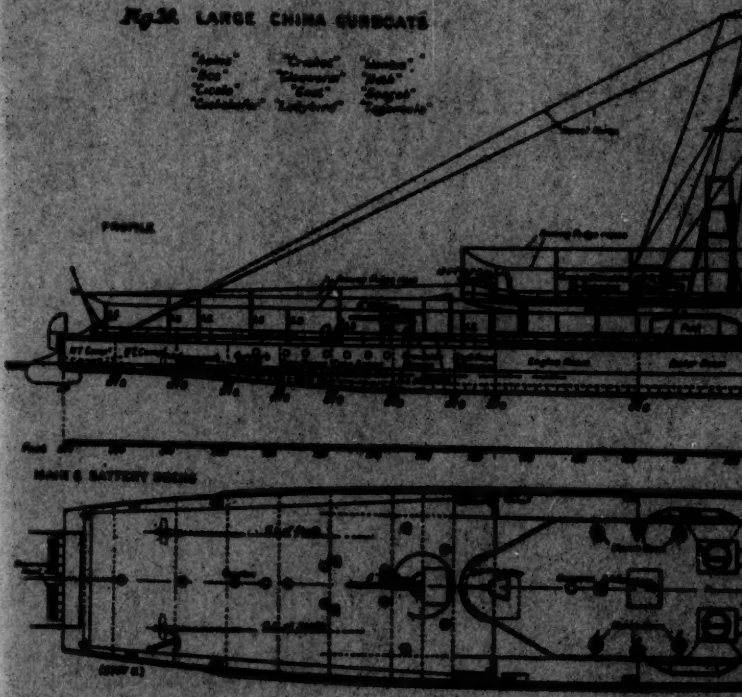
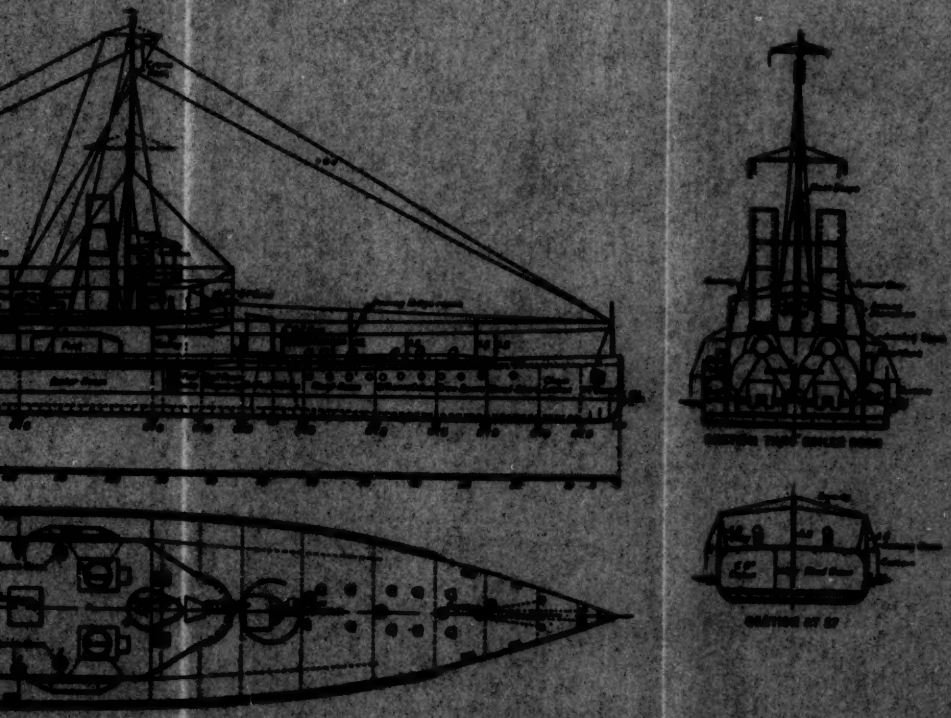

NE SWEEPERS (HUNT CLASS)

Fig 32. LARGE CHINA GUNBOATS





(*a*) The profit-earning capacity of both the small vessels is very low under all freight rates, and the smallest vessel can sometimes be worked only at a loss.

(*b*) The profit-earning capacity in the longest vessel is not so good as in the 490-foot vessel for a given price of coal and given freight rate, and the 490-foot vessel is the all-round best one for this voyage.

(*c*) With a 7*s*. 6*d*. freight rate, the speed of this vessel should never exceed 10 knots; with freight at 10*s*. it should remain at 10 knots for coal at 25*s*. per ton, and rise to 13 knots with coal at 12*s*. per ton; and with freight at 20*s*. it should be at least 14 knots, unless coal is more than 26*s*. 10*d*. per ton.

(*d*) The largest vessel is really suffering badly from the long time taken to discharge and load cargo. This time is taken from Mr. Anderson's paper, and it was pointed out in the discussion on the paper that this was high. If n_1 is reduced, R is increased, and the maximum profit-earning will also be increased.

(*e*) It will be understood that for shorter or longer voyages the results may be different. This can be seen by varying n_1 in equation (3) for R. A shorter journey increases the best speed for a fixed price of coal, but the amount would depend upon several circumstances.

(*f*) By putting $2\,R = \frac{yP}{365}$ the condition for working at no profit is obtained, R being defined here by formula (3).

§8. The cost factors *x*, *t*, *y* and P are all known with fair accuracy for vessels employed on fairly regular trade, and although *q*, *t* and P are liable to vary, they are quite definite, and it is a comparatively easy thing for any designer to find the limiting speed appropriate to average limiting values of freight rate *f* and coal value *q*. The vessel should be designed to obtain the highest speed appropriate to these average values. With a good form the propulsive efficiency will then always be good at lower speeds, and good profit returns will result. If the design is arranged for average speed or average freight rate and coal price, the vessel may lose some or all of the advantage of any rise in freight above the average, because of its bad propulsive performance when the speed is increased.

§9. In all calculations of this kind, a number of assumptions are made, and some known factors are neglected, and due weight should be given to these when selecting the speed for any voyage or the displacement and speed for any trade route. For example, it is known that:

(1) The larger a vessel the more seaworthy it can be made, and the less is its speed affected by bad weather.

(2) Reserve coal to meet bad weather must be carried, and the proportion which this reserve bears to the whole decreases both with speed and with displacement.

(3) Relatively greater facilities are required for handling cargo in large vessels than in smaller ones.

(4) Greater coaling facilities are required for large vessels.

It has been assumed that all the vessels on each voyage have full cargoes. Loss of cargo is equivalent to smaller *r*, and therefore reduces R, *i.e.*, tons of coal burnt per day, which means running at less than full profit. It would be necessary to work this out in detail to tell whether the large or small vessel is most affected, but presumably the larger one is most liable to run under these conditions, as it deals with larger weight of cargo.

(5) The assumption that the indicated horsepower varies, as the cube of the speed does not hold good for many cargo vessels now built, or for any ordinary cargo vessel if it is forced to speeds in the region of 14 knots (on a length of 400 feet). The formula will in this case overrate the proper speed to a certain extent.

§10. *Ship Dimensions.*—It will be seen from the foregoing that actual dimensions do not enter into the problem of the earning capacity of a ship, except as they affect deadweight, displacement and economy of propulsion. Two at least of the principal ship dimensions, namely, length and draught, are so involved with port facilities that in many cases these facilities, and not propulsion at all, define what is possible with both length and draught, and the naval architect's usual problem is to produce the least resistful form of given displacement without altering these dimensions. At this stage it would be useful to state the various forms of resistance which a cargo vessel experiences and their relative magnitude. At quite low speed there is a small amount of wave making, a little eddy formation, and a great deal of skin friction. The latter amounts to about 80 per cent of the whole. At higher speeds, *i.e.*, at what might be called the limiting practical speed,* the skin friction should be about 70 per cent to 75 per cent of the whole, and the wave making most of the remaining 30 per cent to 25 per cent. But in all tramps there is a small amount of eddy formation on the upper levels at the stern near the body post; this point is dealt with later (§15 and §16). The waves created consist of divergent sets at each end, and transverse waves. Economy in wave making can be affected by so arranging the shape that the bow transverse waves tend to cancel the stern transverse waves at such speeds as these may be important.

§11. *Wetted Surface.*—The smaller the wetted surface per ton of displacement is kept, consistent with non-wave-making, the better will be the propulsive performance. Calculations have shown that this is remarkably non-sensitive to such factors as proportion of parallel body in length, and prismatic coefficient, but depends mainly upon the cross dimensions. With fixed beam, the wetted surface per ton will drop steadily as draught is increased, and a similar drop is obtained if draught is fixed and beam increased. The best ratio of beam to draught is that which gives precisely the same reduction of skin per ton for either beam or draught variation. In Appendix II a formula for this ratio has been obtained in the case of a simplified ship form. This formula has been compared with results obtained by calculation for a number of models whose principal dimensions have been widely varied, and found to give results about 6 per cent high, with parallel body from 25 per cent to 50 per cent of the length. The ratio given by the formula must be regarded as a sort of datum. When the $\frac{\text{beam}}{\text{draught}}$ of a ship is materially less than that given by the formula (corrected for the above 6 per cent), there is something to be gained by increasing beam and decreasing draught. When the displacement is constant and the ratio $\frac{\text{beam}}{\text{draught}}$ is near that obtained by the formula, it may be changed to a quite fair extent without any variation of wetted surface. As a matter of interest, this ratio is about 2.8 for some tank forms of 30 per cent parallel body, and somewhat less than this for forms of 50 per cent parallel body. These proportions require to be borne in mind in settling the shape of midship section.

§12. *Midship Section Area and Shape.*—The displacement and length being fixed, the settlement of these items is the first step in lining out the form. For a given speed and displacement there is a midship section area most appropriate for low wave-making resistance. This area A can be found from the simple formula:

$$A = \frac{6.29 \triangle P2}{V2} \text{ square feet.}$$

* i.e., the highest speed possible without any abnormal increase in $\frac{\text{resistance.}}{(\text{velocity})}$.

Where V is the speed in knots, Δ is the displacement in tons, and P* is a wave-making constant. Most appropriate limits to the values of P* for low wave-making can be obtained from Fig. 1, in which the unshaded portions give minimum resistance.

Putting these values of P in the above equation, the limiting values of A are obtained.† Experiments on models with different midship-section coefficients‡ point to the employment of the largest possible midship section consistent with the other features of the design in vessels of this type, and this conclusion is supported by the general test work of the Froude tank.

With vertical sides from the load water-line to from 4 feet to 5 feet above the keel, associated with a very small rise of floor (or none at all) and a bilge curve of radius of from 4 feet to 5 feet, a coefficient of 0.98 can be obtained.

From an economical propulsion point of view, a rise of floor has the one advantage of permitting easier bow and buttock lines to be worked when there is a considerable amount of parallel middle body, but if a larger radius of bilge and quite flat floors are used, equally good results can be obtained, and it has been found possible, by careful design of the ship at the sections where the entrance and run join the parallel body, to work a square corner to the midship section without any appreciable loss in effective horsepower. Such a shape abolishes the need for a bilge keel, but may present difficulties in constructional design and in practical use. The shape of section being settled, and the draught being already fixed by port facilities, the beam must be arranged to give an area of section A inside the limits already calculated, or as near as possible to these. It should be noted that, as soon as this is fixed, the prismatic coefficient is similarly fixed and the wave-making characteristics are determined. When it is thought desirable, the aim of this section can be achieved by the tentative process of assuming midship area, length, and prismatic coefficient, and seeing what P value is obtained and how this comes on Fig. 1. If it is in a bad place the assumed data should be adjusted.

Longitudinal Distribution of Displacement.—The next step is to decide upon the most economical longitudinal distribution of displacement which, while giving a reasonable position for the longitudinal C.B. will require the minimum horsepower.

§13. *Parallel Middle Body.*—It is generally advantageous, both from the point of view of space and of low cost of construction, to work as much parallel middle body as possible. To determine the greatest length of perfectly parallel body, consistent with low residuary resistance at the required speed, reference can be made to the papers§ giving results of experiments made at the William Froude National Tank. These papers deal with experiments on models of ships of the average dimensions of tramp steamers, and by choosing the set of experiments on models nearest in prismatic value to that already settled (§12) a very fair estimate can be made of the greatest percentage length of parallel middle body to be safely used for the service speed. Similar experiments carried out by Mr. Taylor support these results, and our work of the last few years shows that they can be used with confidence. For vessels of greater or

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* The origin and meaning of this constant are given in Trans. Institution of Naval Architects, vol. lv, Part II.

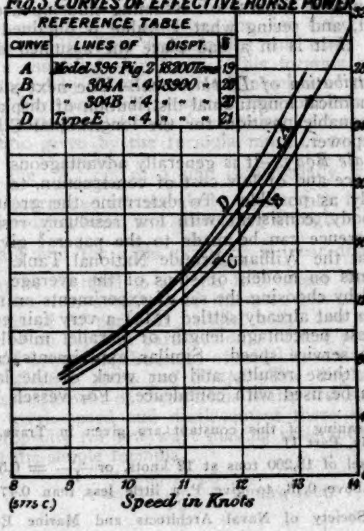
† For a 490-foot vessel of 18,200 tons at 12 knots, or $\frac{V}{\sqrt{L}} = 0.54$ the prismatic coefficient should be above 0.73, to give P a little less than 0.471. That taken for Fig. 2 is 0.77.

‡ Trans. American Society of Naval Architects and Marine Engineers, 1901. D. W. Taylor.

§ Trans. Institution of Naval Architects, vols. lvi and lvii.

Fig. 1. RELATION BETWEEN PRISMATIC COEFFICIENT, LENGTH AND SPEED FOR HIGH & LOW WAVE MAKING RESISTANCE. *The shaded portion indicates high wave making.*

Fig. 3. CURVES OF EFFECTIVE HORSE POWER



(375 c)

less proportion of beam to length than is usual, the results may be slightly different. The result of an investigation of beam effect will be presented at the spring meetings of the Institution of Naval Architects this year. In passing, attention might be drawn to the very large waste of power that may result from a bad selection of length of parallel for a given speed in short full ships. This is very clearly shown in the classical experiments of Dr. William Froude, Transactions of the Institution of Naval Architects, and the Teddington tank parallel-body series "J" and "K" described in the Transactions of the Institution of Naval Architects, vol. lvii.

§14. *Ratio of Length of Entrance to Length of Run.*—When the percentage length of perfectly parallel body has been settled, the relative amounts of the remaining length of the ship which can be put into entrance and run should be determined. Complete experiments† made at the William Froude National Tank, covering a range of prismatic coefficient of 0.85 to 0.66 with 10 per cent, 30 per cent and 50 per cent of parallel middle body, enable a close estimate of the best ratio to use for the particular speed at which the ship is to run, in order to get minimum horsepower. For high proportions of parallel body (50 per cent) an $\frac{\text{entrance}}{\text{run}}$ of about 0.9 is best, and there is a small advantage in working the entrance shorter than the run even with 30 per cent parallel middle body. The prime cause of this advantage in a low ratio of $\frac{\text{entrance}}{\text{run}}$ is the avoidance of eddy making at the stern by the longer run obtained, and provided this is attained, the resistance is not very sensitive to this ratio, and this permits of a certain amount of freedom in adjusting the longitudinal C.B. Tank experiments have shown that for vessels of ordinary proportions the minimum length of run required is given by $4.1 \sqrt{\text{midship section area}}$ (see also §16).

§15. *Prismatic Coefficient of Entrance and Run and Curve of Areas of Sections.*—The designer is now in a position to draw the curve of areas, which will satisfy the length of entrance, run and parallel body, and the prismatic coefficients and displacement already fixed. He will probably prefer to base the first rough shape of the area curve on that of some particular ship not greatly different in dimensions, whose trial results he may have. Guidance as to the best modifications to the form of this curve can be obtained from the published results of model experiments made at the William Froude National Tank and by Taylor and Sadler, in order to determine the effect of change in prismatic coefficient and shape of the ends. The results of the tank experiments can be found in sets F, G and H, Transactions of the Institution of Naval Architects, vols. lvi and lvii. Generally speaking, (*a*) a somewhat full shoulder to the entrance is not detrimental if the speed is low, but a full shoulder to the run may result in an increase in power, due to either eddy or divergent-wave formation. (*b*) The after end of the run should not be cut away to get a fine finish, but the area curve should be carried out with a fair curve to its end, with no marked change in shape just before the body post. (*c*) Every endeavor should be made to avoid a quick change in the slope of the tangent to the curve, and the maximum slope of this tangent to the base should be kept as small as possible. (*d*) When the prismatic coefficient is below 0.7 and the speed above $P = 0.5$ (*i.e.*, 11.2 knots for a 400-foot ship) the curve of areas of sections of entrance should have a little hollow in it, and this should die out altogether for a prismatic coefficient of 0.8 at all speeds.

† Trans. Institution of Naval Architects, vol. lv.

§16. *Load Water-Line.*—For all tramp steamers this should have a slightly convex or straight entrance. The shoulder of the fore end where it runs into the parallel body can be fairly blunt if necessary. The higher the speed required, the finer must the entrance angle be made, and the greater must be the distance from the stem before it turns into the parallel body. At low speeds, say 9 knots for a 400-foot ship, 60 degrees entrance angle (side to side) can be accepted. By straightening or hollowing the load water-lines to produce a finer end, a lower "bow breaker" and smaller divergent waves are formed. Yet it does not follow that the effective horsepower will be lower if the speed is small in comparison with the ship's length. An increase in power of from 4 per cent to 5 per cent may be the result of working a hollow line forward, although at higher speeds (usually beyond the limits of tramp steamers) the influence of the finer end always shows itself in diminished resistance. At the after end, the angle between any tangent to the load water-line and the middle line of the ship, should not exceed about 20 degrees in order to avoid dragging dead water with the vessel. It is, of course, not always possible to keep the line so fine, and should a greater angle than 20 degrees become a necessity, the curve must be so drawn that the maximum slope is as far aft as possible, consistent with finishing at the rudder post without any marked change of curvature or hollow in the curve. It is not generally realized that at low speeds a fine end aft, obtained by filling out in front of it, is quite as conducive to eddy making as a full finish, as the stream lines tend to break up whenever the rate of expansion exceeds that given by a level boundary of 20 degrees to 25 degrees. It is in this connection that the main propulsive advantage of a cruiser stern is obtained. With ships of either considerable beam or high prismatic coefficient, a well immersed cruiser stern enables a reasonable slope to be given the load water-line and the lines immediately below it, and thus eliminates eddy formation in this area, with a consequent reduction of about 3 per cent of the total resistance. The more the feature is immersed, the better is the result obtained.

§17. *Cross-Sections.*—Experiments have shown that there is some advantage in departing from the usual U-sections forward, and adopting sides sloping inward towards the middle line from the load water-line, with well-rounded heels. The displacement lost at the heel of the sections is regained by working the full water-line suggested. This allows the water which is endeavoring to find its way under the form along bow or diagonal lines, to do so with a minimum rate of change of curvature of flow. To attain this same end it is good to commence easing the flat floor of the parallel midship body into the uprising entrance and run sections as soon as the limit of the parallel body has been reached. The area of section can quite easily be maintained by keeping the full beam at load water-line and the levels immediately below it for some distance forward of the parallel-body limit. This same type of section is required where the parallel body merges into the run, and further aft the sections should be of V form, in accordance with Mr. R. E. Froude's dictum. Double curvature should be kept to a minimum. Club-footing the aftermost sections has not been found to exercise any noticeably bad effect on the horsepower.

§18. *Level Lines and Contours.*—There should be a fairly uniform change in the slope of the level lines from load water-line to keel. It is known that, at the fore end, the water flows approximately along bow lines near the keel. These lines will, therefore, give the best indication of the actual stream lines in this neighborhood, and they should be kept as easy as possible for some little distance transversely on each side of the keel. This can be done either by hollowing the lowest levels at their forward extremities, or, preferably, by cutting away the forefoot of the

bow contour. This tends to diminish the "bow breaker," and also decreases the wetted area, and consequently the skin-friction resistance, while sacrificing practically nothing in the way of useful space. The shape of the upper portion of the bow contour has very little effect until wave-making speeds are reached, when forward rake gives slightly longer and easier level and bow lines. At the after end, some hollow is required in the lowest levels to obtain V-shaped transverse sections, but with the early

Extreme or Running Vessels of Various Sizes.

| | | |

middle line, as this gives very steep buttocks. Above the propeller aperture full advantage should be taken of the extra length of the levels which now run to the rudder post. Filling out these levels prevents any eddy formation in this area, and effects an appreciable reduction of resistance in the same way as does a cruiser stern.

§19. To illustrate this portion of the paper a design for a 490-foot vessel of 18,200 tons (see Appendix I) has been drawn out, and is given in Fig. 2. This vessel is required to steam at a maximum speed of 14 knots when the freight rate is high, and at 10 knots when freight is low and coal high, Appendix I, &c. The prismatic coefficient taken is 0.77. This evades wave-making up to about 12.3 knots P = 0.471, and gives a midship area of 1,670 square feet, corresponding to a beam of 62.75 feet and draught of 26.83 feet, the rise of floor being nil. Of the total length 30 per cent has been made perfectly parallel middle body. The ratio of length of entrance to run has been taken = 0.75, this gives a length of run = 196 feet, and entrance 147 feet. The angle of entrance of the load water-line has been fixed at 55 degrees, and all the bow sections cut away at the heel, the curve of areas of sections having only a very slight hollow in it. This cutting away has involved the contour of the stem being raised to avoid hollow lines. At the stern the load water-line is 29.5 degrees. The sections have been lifted at the heels as soon as possible aft of the parallel body, and the form has been filled out above the propeller aperture. The curve of effective horsepower for this form is given in Fig. 3. It should be noted that this design is given to illustrate general principles, and that careful consideration is required before it is adopted for conditions other than those assumed. For example, its propulsive efficiency is falling off at 14 knots and rapidly diminishes at higher speeds.

§20. *Straight-Frame Ships.*—This type of vessel only differs from that considered in the preceding paragraphs in having the sections built up of straight lines instead of curves. The same considerations apply with little qualification to this type as to the other. One additional matter, however, required attention, namely, the run and number of the chine or knuckle lines. A variety of forms of this type have been tested at the tank to the order of the Deputy-Controller of Auxiliary Shipbuilding, and the Controller-General of Merchant Shipbuilding, and the authors are indebted to Lord Pirrie for permission to publish the following data derived mainly from such experimental work. The run of the chine, particularly in elevation, is a most important factor. This can be so adjusted that the resistance for the form is within a few per cent of that of the curved section ship. With a single chine form, *i.e.*, one in which the side frames and floors meet at a sharp angle with no attempt to bevel or round the joint, this excess resistance can be reduced to about 6 per cent, or with allowance for bilge keels, which are not necessary on the single chine form, to 3 per cent or 4 per cent. With the chine bevelled away throughout the form, giving a double-chine line, a slightly better result can be obtained. This excess resistance is due to a small amount of eddy formation at the chine. Several single-chine forms have been tested with the sharp angle of the sections replaced by curves of 4-foot radius throughout, and these gave a performance of the same as for the ordinary curved-section type. The lines for a vessel of this type are given in Fig. 4 and the results in Fig. 3. A form similar to this, with sides carried down to the keel line, and having a sharp angle at the joint, gave results some 60 per cent worse. Lifting this knuckle to an increasing amount towards the ends steadily improved results, until a line, as given in Fig. 4, was reached. It was also found that, as for the curved-frame ship, that the rise of floor should increase as soon as possible in entrance and run, but the sides may be kept to full or water-line beam for some little distance into entrance and run.

Fig. 2.

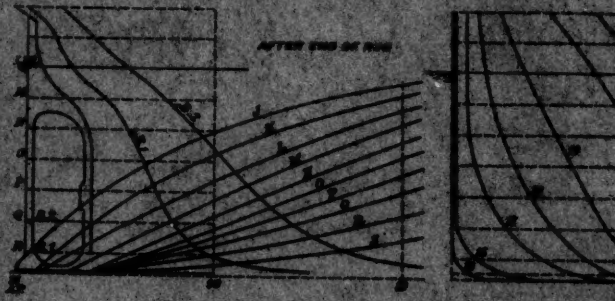
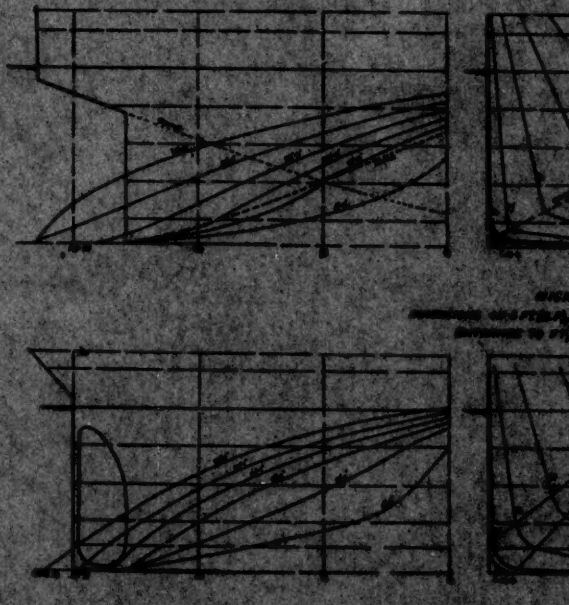
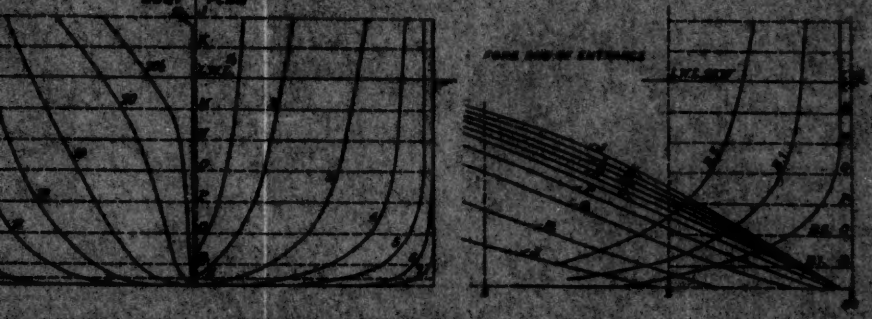
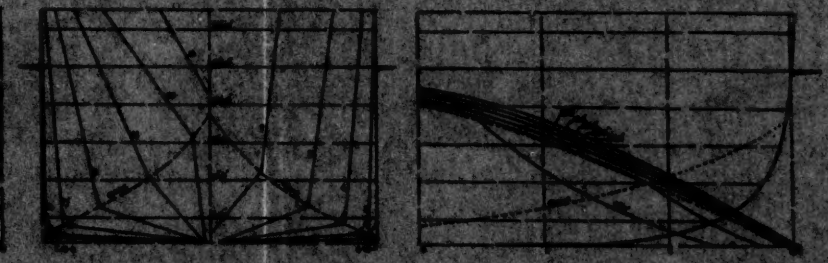
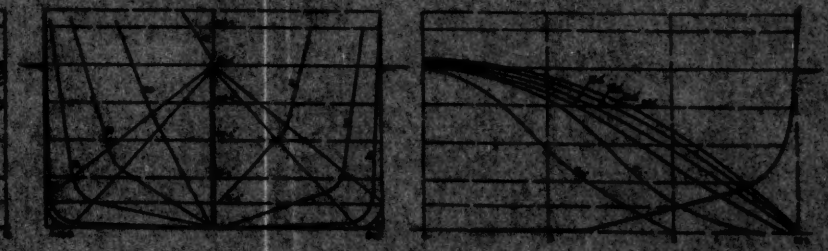


Fig. 4.









§21. Similar tests have been made with models in the American tanks, and a number of results have been published. In Fig. 3 is reproduced the effective horsepower curve for the Michigan tank, type E,* a model of approximately the same dimensions and displacement as Model 304. The lines of this model are given in Fig. 4. A form of about the same relative dimensions, but of a different type (developed in the Washington tank) having Y after sections and marked U sections with vertical sides forward, gave results very similar in power to the Michigan model, being slightly worse at 8 knots and 2 per cent or 3 per cent better at 12 knots.

APPENDIX I.

To find the best speed and displacement for a voyage of 4,000 miles, when the freight rate is 20*s.*, 10*s.*, and 7*s.* 6*d.* per ton, capital charges as insurance, depreciation, repairs, &c., per year being one-seventh initial cost ($y = 1/7$), loading, discharge, brokerage and management charges 3*s.* 6*d.* per ton ($t = 0.175$ pound), tonnage dues 1*s.* 3*d.* per ton. Delays for coaling, repairs, holidays, &c., and coal consumed taken as in Mr. Anderson's paper.

For the purpose of the example, the calculation has been made for the five vessels, as used in Table I of Mr. Anderson's paper, and as a result the table is much larger than would ordinarily be required.

Cargo deadweight C has been taken constant for all speeds, but actually would be a little less at high speeds than at low. This could easily be set right in an actual ship, but it makes little difference in comparing different ships, and if necessary could be corrected by slight adjustment of the M value in the last line.

It should be noticed that even fair changes in such items as time for loading and repairs, &c., will not make a serious change in comparative results, *i.e.*, they only want fixing approximately.

For further remarks on the table see §8, and for definition of symbols see §4.

APPENDIX II.

Best ratio of beam to draught for minimum wetted surface per ton displacement, for solid of geometrical form.

The solid is shown in plan in the figure, and is wall-sided of draught "a" feet.

The displacement $\Delta = 2\,a\,b\,(l + c)$ the wetted surface "S" $= 2\,(a + b)\,l + 2\,b\,c + 4\,a\,\sqrt{b^2 + c^2}$. The best dimensions are those for which $\frac{d}{d\Delta}\frac{S}{\Delta}$ is the same for change of Δ either by beam or draught variation.

* This is the best form, for which complete data were published by Mr. Saunders in August, 1917. Professor Sadler has published (in November, 1918) some results for a form Y i c, which appears to be better in performance, but the data are not sufficiently complete to enable a definite statement to be made.

For a change $\frac{d\frac{S}{\Delta}}{d\Delta} = -\frac{1}{2a^3b(l+c)^2}$

for b change $\frac{d\frac{S}{d\Delta}}{d\Delta} = \frac{4b^2}{\sqrt{b^2+c^2}} - 4\sqrt{b^2+c^2} - 2l$

$$4a^3b^2(l+c)^2$$

and for these to be equal

$$\frac{b}{a} = \frac{2\sqrt{b^2+c^2}+l-\frac{2b^2}{\sqrt{b^2+c^2}}}{(l+c)} = \frac{2c^2+l\sqrt{b^2+c^2}}{(l+c)\sqrt{b^2+c^2}}.$$

For a box, *i. e.*, blunt ends, $\frac{\text{full beam}}{\text{draught}} = 2{\cdot}0.$

For a form with no parallel body $\frac{\text{full beam}}{\text{draught}} = 4$

$$\left\{1-\frac{1}{4}\left(\frac{\text{beam}}{\text{length}}\right)^2\right\}$$

TERRY REDUCTION GEARS.

Terry reduction gears manufactured by The Terry Steam Turbine Company of Hartford, Conn., are again on the market, not having been obtainable during the last year, due to the concentration of this company almost entirely on turbines for the Destroyers. Although made primarily for sale with *Terry* turbines, the gears alone are available whenever a high-grade reduction gear is desired.

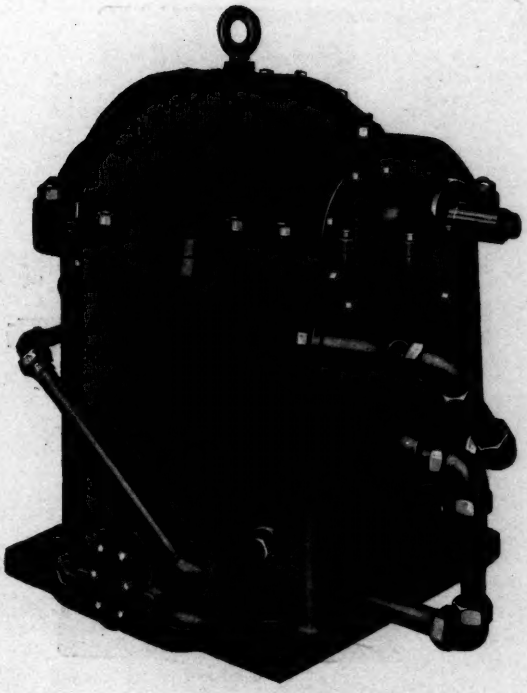
In the design of *Terry* gears the same qualities which made the *Terry* turbine so popular, namely, simplicity, reliability and interchangeability, have been incorporated. There are a number of features of design which are particularly interesting.

The *Terry* gears and pinion are of the stub-tooth, double-helical type, generated to true form on a Fellows Gear Shaper. The accuracy of this method of tooth generation is such that no grinding or scraping process is necessary to insure perfect contact of tooth surfaces. The gears and pinions are interchangeable, an advantage not often found where the final finish of the teeth is by grinding or scraping.

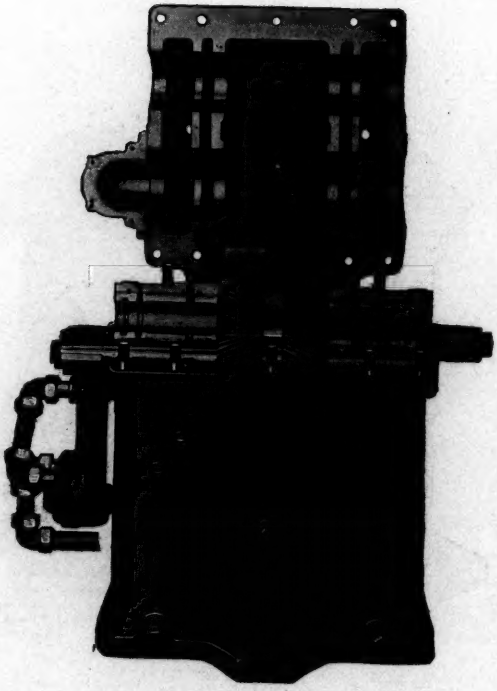
A well ribbed, double-walled, box-like structure, extending the full depth of the case, forms a rigid support for each pair of bearings. The space between the walls acts as a water jacket for cooling the oil. The ribs between the walls act both as stiffening members and water baffles. The central part of the case, directly under the gears, forms an oil reservoir which contains sufficient oil to supply not only the gears but also the turbine.

The case being of the double-walled, rigid, box-like structure, accurate maintained alignment of the gears is insured. The casing is bored with the aid of fixtures, the dimensions and alignment of the boring being checked by accurate gages. This results in a tight and rigid fit of the bearings.

The bearings are of ample size, insuring safe pressures and rubbing speeds. They are split horizontally to permit their replacement without removing the couplings. The bearing shell is of cast iron and, as can be

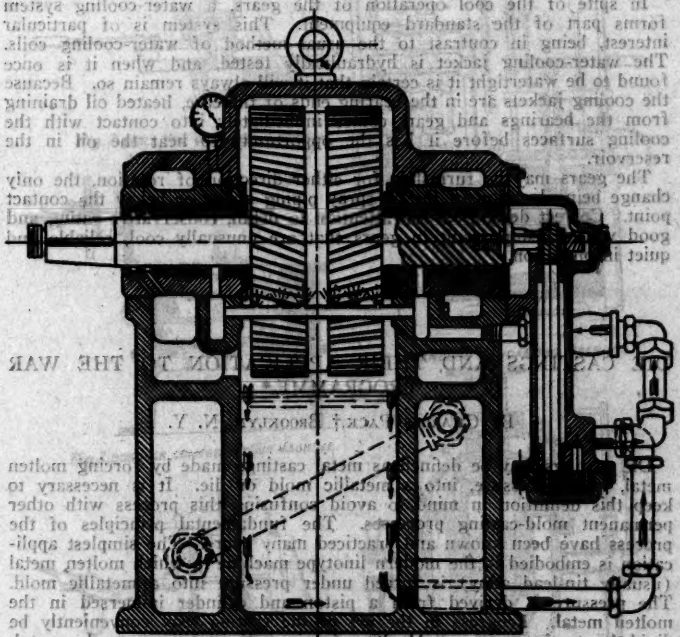


TERRY GEAR CASE.



TERRY GEARS, COVER LIFTED.

seen in the cross section, is unusually heavy. This is to prevent warping or "pinching-in" from heating, which so often occurs in thinner shells of bronze or similar metal. The lining used is of highest-grade tin base babbitt on both gear and pinion. The bearings are accurately machined to close limits and require very little scraping. Their interchangeability makes it possible, in case of excessive wear, to easily and quickly replace them and restore the original alignment.



Cross Section Terry Gears

The oiling system employed is particularly interesting. It is a forced-feed system, the ring-oiling system having been found unsatisfactory for turbine reduction-gear bearings. By reference to the sectional view it will be noted that the oil pump is located well below the oil level in the reservoir, to avoid suction lift, thereby preventing ruin of the gears due to possible failure of the pump to function. The oil is pumped from the reservoir through short, direct, brass piping to a self-cleaning strainer, thence through distributing passages to large, annular oil pockets around each bearing shell, and through the spray pipe from which the oil is sprayed, for lubrication of the gear teeth. The oil-pressure gage is located in one of the above-mentioned annular oil pockets at the most distant point from the oil pump.

The pump and its bevel-gear drive make a complete unit without stuffing boxes or exposed running parts. The unit is so constructed as to be easily accessible. The pump gears may be removed for inspection without disturbing the driving mechanism or oil piping. The bevel gears may also be inspected by removing a small cover. The effectiveness of this lubricating system has been proved by test for, without any water cooling, the temperature of the oil after long runs is less than that found in most gears when running with water cooling.

In spite of the cool operation of the gears, a water-cooling system forms part of the standard equipment. This system is of particular interest, being in contrast to the usual method of water-cooling coils. The water-cooling jacket is hydraulically tested, and when it is once found to be watertight it is certain that it will always remain so. Because the cooling jackets are in the bearing ends of the case, heated oil draining from the bearings and gears comes immediately into contact with the cooling surfaces before it has the opportunity to heat the oil in the reservoir.

The gears may be furnished for either direction of rotation, the only change being location of the oil-spray piping above or below the contact point. Correct design, careful attention to detail, conservative rating and good workmanship result in gears that are unusually cool, reliable and quiet in operation.

DIE CASTINGS AND THEIR APPLICATION TO THE WAR PROGRAMME.*

By Charles Pack,† Brooklyn, N. Y.

Die castings may be defined as metal castings made by forcing molten metal, under pressure, into a metallic mold or die. It is necessary to keep this definition in mind to avoid confusing this process with other permanent mold-casting processes. The fundamental principles of the process have been known and practiced many years. The simplest application is embodied in the modern linotype machine in which molten metal (usually tin-lead alloy) is forced under pressure into a metallic mold. The pressure is derived from a piston and cylinder immersed in the molten metal. Progress in the art of die casting may conveniently be divided into three groups: Machine for imparting pressure to the metal, material for the die or mold, casting alloys.

Casting Machines.—The problem of delivering molten metal under pressure into a die is comparatively simple, when dealing with low fusing-point alloys, as the alloys of lead and tin, but it is much more complicated when dealing with metals of higher fusing points, such as the alloys of zinc, aluminum and copper. Although the art of die casting is comparatively new and, to a large extent, unknown, the records of the Patent Office are replete with patents on the subject.

Fig. 1 shows the Underwood machine patented in 1902; this is probably one of the first machines designed for the production of commercial die castings. The relation of this machine to the linotype-casting machine is clearly apparent. A cylinder and piston are immersed in the molten metal, the application of power to this piston forcing the molten metal, under pressure, into the mold or die. The Doehler machine, Fig. 2,

* Paper read before the American Institute of Mining Engineers, February, 1919.
† Chief Chemist, Doehler Die Casting Company.

Fig. 1. UNDERWOOD DIE-CASTING MACHINE

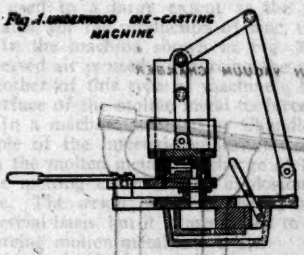


Fig. 2. EARLY TYPE OF DOEHLER MACHINE

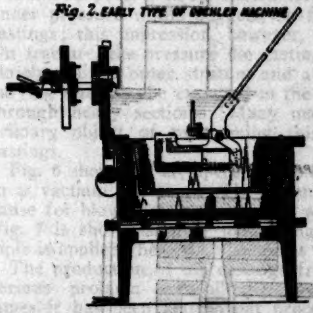


Fig. 3. DOEHLER COMPRESSED-AIR MACHINE.

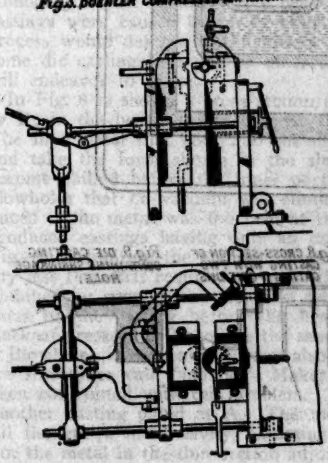


Fig. 4. ANOTHER TYPE OF COMPRESSED-AIR MACHINE.

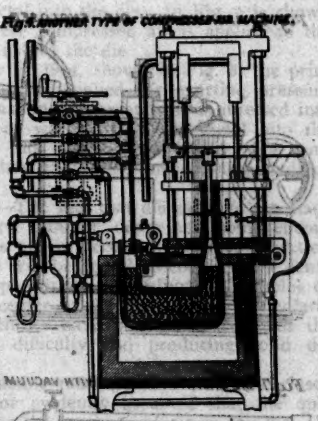


Fig. 5. CHANDLER DIE-CASTING MACHINE

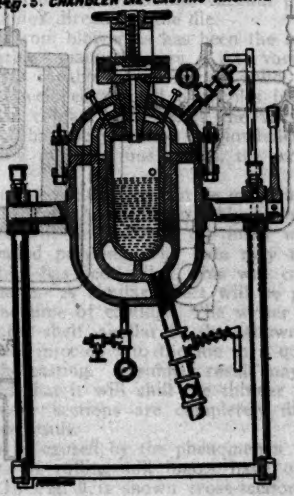


Fig. 6. MACHINE WITH DIE ENCLOSED IN VACUUM CHAMBER

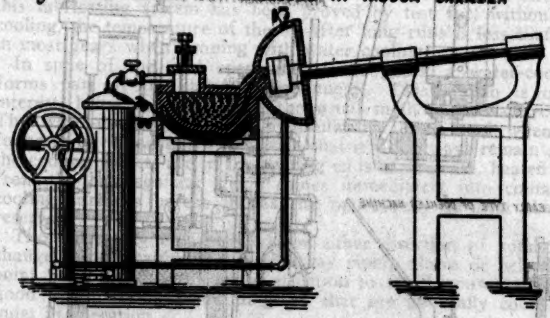


Fig. 7. DIE-CASTING MACHINE WITH VACUUM APPLIED DIRECTLY TO DIE.

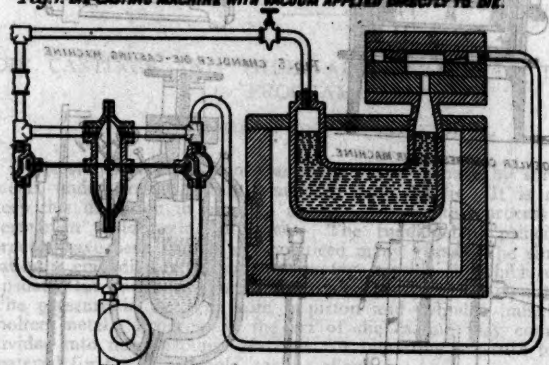
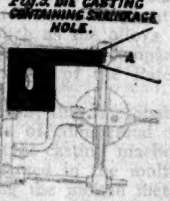


Fig. 8. CROSS-SECTION OF CASTING WITH TWO GATING PORTIONS.

Fig. 9. DIE CASTING CONTAINING SHRINKAGE HOLE.



patented in 1907, is based on the same general principles. This machine is used to a large extent at the present time, throughout the United States, for the production of zinc, tin and lead alloy die castings.

In the machine shown in Fig. 3, patented by Doehler in 1910, compressed air is used for forcing the metal into the die. In Fig. 4 is shown another of this type of machine. Here compressed air is applied to the surface of the molten metal to force it into the die.

In a machine patented by Chandler in 1914, shown in Fig. 5, the principle of the internal-combustion engine is applied for exerting pressure on the molten metal. A charge of gasoline vapor and air is injected into the melting chamber, the explosion of which forces the metal into the die. The writer has never heard of this machine being used on a commercial basis, but it is mentioned to show the various means suggested for forcing molten metal into a die.

Methods Used to avoid Blowholes.—The fact that die castings are made under pressure would suggest, on first thought, dense and homogeneous castings; this impression, however, is not in accord with actual practice. On fracture, the pressure die casting will be found to consist of a dense closely-grained outer stratum and a porous inner stratum. Blowholes of varying size may be expected in the center of the die casting, particularly through heavy sections. Many machines have been designed with the primary object of overcoming this difficulty and producing solid die castings.

Fig. 6 shows an air-operated die-casting machine with the die enclosed in a vacuum chamber. The inventor evidently assumed that the only cause for blowholes in the casting was the presence of air in the die. In Fig. 7 is shown another die-casting machine in which the vacuum principle is applied; here the vacuum is applied directly to the die.

The production of die castings free from blowholes has been the most serious problem controlling die-casting manufacturers. At various times it has been stated that processes capable of producing solid and homogeneous die castings have been developed. If all blowholes in die castings were caused by air coming in contact with metal, the vacuum process would deserve consideration. That the presence of blowholes in some die castings are due to other and more serious causes, the writer will endeavor to prove.

In Fig. 8 is shown a cross-section of a casting that can be gated at A or B; in the best foundry practice the gate A would probably be used. The first metal that goes into the die will chill around the inner walls and take the form shown in the shaded portion. The gate may then become chilled before the inner portion has been filled; this will cause blowholes that no vacuum will eliminate. A similar effect will be produced if the metal was too cold at the time of casting. The writer has produced castings having only an outer shell, similar to that shown in Fig. 8, by limiting the amount of metal injected into the die to a quantity less than that required to make the casting. A similar result may be obtained by running the metal so cold that it will chill the thinner sections of the casting before the heavier sections are completely filled. Lack of pressure will produce the same result.

Blowholes in die castings may also be caused by the phenomenon that we sometimes call "piping." Makers of rolling-mill ingots have often been confronted with this problem. In Fig. 9 is shown cross-section of another casting gated at A. The metal flowing into the die at A will fill the entire mold cavity, assuming all casting conditions to be ideal, but the metal in the thin section adjoining A will chill before the heavier section, so that, the chilling being from the outside, a shrinkage hole will be left in the center. Here, again, no advantage can be gained by the use of the vacuum system.

Dies.—In the manufacture of die castings from zinc, tin and lead alloys, dies made from low-carbon machine steel last almost indefinitely and answer every purpose. In the first attempts to die-cast aluminum, the problem of obtaining a suitable die material presented serious difficulties, which were described by the writer in a paper read before the American Institute of Metals in 1915. This problem, however, has been solved by the use of various alloy steels making possible the commercial die casting of aluminum and its alloys, which constitute the greater part of the die-casting industry of today. The proper gating and venting of these dies are problems that arise daily and on the solution of these problems depends the success or failure of the process.

Alloys.—In a paper read before the American Institute of Metals in 1914, the writer described the various types of zinc, tin and lead alloys used in the die-casting process. The application of these alloys and their limitations were also pointed out. At that time the die casting of aluminum and its alloys was barely beyond the experimental stage. During the past four years the most important advance in the art of die casting has been made in the perfection of the process for die-casting aluminum and its alloys. The importance of this achievement as an aid to winning the war is best demonstrated by the fact that at least 95 per cent of the die-cast parts used directly or indirectly as materials of war were made from an aluminum-base alloy. Of these castings, only a very small percentage could have been produced successfully in 1914.

Investigations of the casting properties of metals and alloys in the past have been generally limited to sand castings; few data are available as to the casting properties of metals or alloys in metallic molds. Just what constitutes a good die-casting alloy is a subject of unusual interest. A few of the important requirements, outside of the usual physical properties demanded of alloys, are:

Melting-Point.—The successful die-casting machine in every instance is constructed of iron, in one form or another. The melting-point of the alloy must be such that it will melt readily in an iron pot.

Solvent Action.—The solvent action of the alloy on iron must not be too great. Molten aluminum dissolves iron very rapidly and analyses of aluminum die castings on the market will show an iron content of from 1 per cent to 3 per cent, due to the solvent action. Fortunately there is no serious objection to the presence of iron in aluminum casting alloys. Should the aluminum absorb much above 3 per cent iron, the melting point becomes too high and the alloy becomes viscous and unsuitable for making castings.

Elongation.—The elongation of the metal is of vital importance in determining the die-casting properties of an alloy. Not only is it desirable to know the elongation of the alloy when cold, but it is of greater importance to determine the elongation at various temperatures ranging from the melting-point of the alloy down to normal temperature. The reason for this becomes apparent when the physical phenomena of the die-casting process are considered. Let us assume that a ring, 12 inches (30.48 cm.) in diameter is to be die-cast in a metallic mold around a metallic core. As the molten metal strikes the mold it solidifies. Here a change of state occurs that is accompanied by a reduction in volume, commonly termed shrinkage. Unlike a sand core, the metallic core is not compressible, and retains its original size and form so that the shrinkage of the metal is converted into a stretching action on the solidified casting. If the elongation of the alloy at that temperature is not high enough to withstand this stress the casting will crack. In the usual die-casting practice it is not practical to remove the casting from the die at the solidification temperature of the alloy. For example, the solidification temperature of the aluminum-copper alloys used in the die-casting process is approxi-

mately 1,150 degrees F. (621 degrees C.). It has not been found practical to run the casting dies above a temperature of 500 degrees F. (260 degrees C.), which means that the castings are withdrawn from the dies at that temperature. It follows that the casting is subjected to another stretching stress after the casting has solidified, due to the contraction in volume that must occur when a casting is cooled from a temperature of 1,150 degrees F. to 500 degrees F.

The writer has been unable to find any reliable method for determining quantitatively the elongation of alloys at various temperatures. Many methods have been suggested, but they have proved of doubtful value. The simplest way is to use the old "try-and-see" method. To test the alloy, a casting is made in a die having a comparatively large core and thin wall. If the alloy can stand the casting stress, a perfect casting will be obtained, otherwise the casting will show bad cracks. Only a comparative result is obtained, but for everyday control it answers the purpose. However, a simple and reliable method for determining quantitatively the elongation of metals and alloys at various temperatures would prove of enormous value to all metallurgists engaged in the various phases of metal-casting research.

It is interesting to note that the elongation of a metal or alloy at normal temperatures is no indication as to the properties of that metal or alloy at higher temperatures. The writer has found many cases where an alloy showing little or no elongation at normal temperatures shows a high elongation at higher temperatures. The alloys of aluminum and copper may serve to illustrate this point. It is well known that the addition of copper to aluminum reduces the elongation of the aluminum alloy. An aluminum alloy containing 12 per cent copper will show less elongation than an alloy containing only 6 per cent copper when tetsed at normal temperature. Nevertheless, the 12 per cent copper alloy has a greater elongation at higher temperatures than the 6 per cent alloy, and consequently the 12 per cent alloy is better able to withstand the casting stresses to which it is subjected in the die-casting process.

In the early days of the die-casting industry alloys were compounded indiscriminately, and little or no consideration was given to the metallurgical principles involved. The manufacturer in many instances knew much more about machinery than about metals. The result was that there were put on the market die castings made from alloys that deteriorated rapidly and created a prejudice among engineers against the use of these castings. That this prejudice was in part justified must be admitted; nevertheless, the modern die-casting plant is equipped with physical and chemical testing laboratories and the die-casting practice of today bears no relation to that of five years ago.

Die Castings Made for War Purposes.—Die castings have had their most severe test during the past two years, during which time most of the die castings manufactured were used directly, or indirectly, in the Government's war program. Here is a partial list of the application of die castings for this purpose:

Gas masks, breather tubes and other metal parts.

Lewis machine guns, 100 die-cast parts to every gun.

Browning machine guns, four of the most vital parts.

Naval and army binoculars, the entire housing.

Army truck, tank and airplane die-cast parts include parts of ignition system, carburettor gasoline-regulating devices, steering-wheel accessories, ball-bearing cages bearings, speed indicators, &c.

Pistol, complete signal pistol.

Submersible bombs, some designs contained as many as 10 die-cast parts.

Hand and rifle grenades, every grenade manufactured in this country contained one or more die castings.

Trench mortar shells, plugs die cast.
Airplane drop bombs, one or more die-cast parts.
Surgical instruments, including hair clippers, respiratory devices, &c.

In many instances, die-cast parts were used where the failure of the part would result in serious loss of life. The fact that not one failure of a die casting has been reported must continue to be a source of deep satisfaction to the modern die-casting manufacturer.—"Engineering."

ON THE METALLURGICAL INFORMATION REQUIRED BY ENGINEERS.*

By Lieut.-Colonel C. F. Jenkin, R. A. F., Member.

For the past four years the writer has been responsible for recommending materials for all parts of aeroplanes and aero-engines. In engineering it has been customary to adhere to the use of materials which experience has shown to be suitable for the different parts, but in aircraft a new factor is of paramount importance—viz.: weight—with the result that new materials, such, for example, as aluminum alloys, are constantly being suggested and tried for purposes for which they have never before been used. Further, it has become of importance to use all materials in their strongest condition; thus the bulk of the steel used is so treated as to have a strength about double that of the steel used for ordinary engineering work. In order to be able to judge whether a new material—or an old material in a new condition—is suitable for any particular use the following fundamental problem therefore becomes of essential importance: "On what properties of the material does its suitability for any particular purpose depend?"

This question has arisen in connection with all sorts of materials—metals, timber, flax and hemp, rubber, paint, &c., and in connection with their use in engines, aeroplanes, balloons, armament, instruments, &c. To simplify the problem let us confine our attention to metals for use in engines. In order that a metal may be suitable for a particular part of an engine it must possess certain properties which the engineer can easily specify, *e.g.*:

1. Stability, *i.e.*, strength to resist deformation—beyond a minute extent.
2. Long life, *i.e.*, endurance when subjected to stress—of a definite sort—and when subjected to abrasion and corrosion.
3. Special properties for certain parts, *e.g.*, a low coefficient of friction for bearing metals.

For simplicity we may confine our attention to items 1 and 2. Stability and long life, then, are the properties desired by the engineer; but these are not among the properties which are known to the metallurgist, and when the engineer asks for information he is given a very different list of data, viz.: Chemical analysis; microstructure; mechanical tests—ultimate strength, elastic limit, elongation, reduction of area; bend tests of different sorts; Brinell, scleroscope; Izod; fatigue range; chemical reactions.

There is hardly any correspondence between the two lists. In an attempt to bridge the gap between the two we use a terrible list of vague words to hide our ignorance—Hardness; springness; toughness and brittleness; fibrous or crystalline structure; resistance to shock or vibration or impact.

Some comparison may be attempted between the two lists by putting them in parallel columns:

* Note read before the Institute of Metals, March 20th, 1919.

Stability under tensile or compressive stresses	Elastic limit.
Stability under bearing pressure (rivet holes, pin holes, ball bearings)	Scleroscopic hardness. Brinell hardness. Resistance to scratching.
Life under steady stress	Elastic limit.
Life under varying stress	Fatigue range.
Life under abrasion	"Hardness" (one of the meanings).
Life under corrosion	The chemical reactions on which corrosion depends are little understood.
Of no interest to the engineer	Chemical analysis.
Of no interest to the engineer	Microstructure.
Little interest except as giving a margin of strength before final fracture in some steadily stressed parts	Ultimate strength.
No interest	Elongation.
No interest	Reduction of area.
No interest	Bend test.
No interest	Brinell.
No interest	Izod.

To illustrate the want of correspondence between the two columns, let us suppose that it is desired to replace steel by an aluminum alloy for some purpose. It is obvious that it is impossible to make any useful comparison by comparing the analyses or the microstructure of the two metals. It is really useless to compare the ultimate strengths, elongations, reductions of area, Brinell, and Izod figures. For consider the elongation. Why do we specify 17 per cent elongation for a steel crank shaft when 1/10 per cent, if it occurred, would ruin it? Is there any reason, then, to specify 17 per cent elongation for an aluminum crankshaft? These apparently important mechanical tests are really no more rational guides than, say, color. As a matter of fact, an experienced eye can judge as well by the color of the steel fracture as by the Izod figure of a fracture. But it is obviously useless to compare steel and aluminum by color.

If the metallurgist's properties are so useless to the engineer, why are so many of them included in the specification? Owing to the lack of knowledge about the life and stability of materials it is not possible to specify such qualities, and the engineer has to specify some definite material which he wishes to use, either for an experiment or as a result of former experience. For this purpose he uses the chemical analysis to define the material, and the microstructure and mechanical tests to define the condition of the material.

What, then, is needed to enable a satisfactory answer to the problem to be given?

1. It is necessary to ascertain what physical properties of the metal they are which determine its behavior under the conditions specified by the engineer.

2. It is necessary to determine tests which shall show the extent to which those physical properties exist in any metal.

3. The names for these properties must be accurately defined.

As an example, consider a crank shaft. It is suggested that the only properties which may need to be considered are:

1. The fatigue range. The designed stress must not exceed this.

2. Freedom from sharp cracks or faults. Such faults may be due to casting flaws or to a crystalline structure which encourages surface cracks during machining.

Wearing property may be found to be unnecessary with properly lubricated bearings—if not, this must be added to the list.

All the other properties enumerated on the previous pages may be found to have no direct bearing on the suitability of a material for crankshafts. It may be gratifying to a steel maker to see a broken crankshaft twisted into all sorts of shapes after an accident, but why should it not have snapped off like glass?

The above two requirements for a crankshaft may be referred to more elementary properties by the metallurgist. He may be able to indicate the types of solution or crystalline aggregate which are likely to give the desired results. If we could arrive at such a result it would be a great advance on our present position. Such complete solutions have been reached in a few cases, and are certainly not impossible.

As an example of a test properly designed to determine the required property, Strohmeyer's test for the fatigue range may be given. A little further research on this test will probably make it perfectly trustworthy. There is no doubt what it measures.

A satisfactory answer to the problem should not be beyond the reach of the metallurgist and engineer if they work in conjunction. The reason that former attempts have had such relatively unsatisfactory results appears to be that engineers and metallurgists have not worked together, and that engineers have not fully realized the fundamental nature of the problems they were attacking.

It may be possible to correlate the essential engineering properties with the results given by existing types of test, and it is on these lines that most work has hitherto been done, but more probably new methods of testing will have to be devised. If new methods of testing are needed it is important that they should be designed to give the precise information required. Too many new tests have been devised in recent years without any exact knowledge of what their results signify. As a consequence a great amount of research has been expended in endeavoring to interpret the meaning of the new tests, which research would have been more profitably spent in devising tests to give the essential data.—"The Engineer."

INSTRUCTIONS FOR SAFE USE OF PULVERIZED FUEL.

Pulverized coal fuel involves certain elements of danger due to inflammable and explosive properties, as is the case also with oil and gas fuel. In order to ensure efficient service and to avoid accidents, therefore, it is necessary to provide careful handling by men properly instructed as to the conditions. Since accidents might tend to bring this comparatively modern system of fuel into disrepute a set of rules and instructions relating to the preparation, storage and use of pulverized coal have been prepared by the Pulverized Fuel Equipment Company, of New York, United States. These rules are given below, with some supplementary information from another company.

GENERAL INSTRUCTIONS FOR FUEL PREPARING, STORING, AND DISTRIBUTING EQUIPMENT.

NOTE.—Stop all fuel leaks.

1. The operation of pulverized fuel drying, grinding, distributing and storage plants should be under the supervision of employees who are

fully instructed and informed with respect to the proper care, upkeep and operation of all the equipment in conformity with the Pulverized Fuel Equipment Corporation's rules and regulations for the preparation, handling and burning of pulverized fuel.

2. Pulverized fuel in suspension is, under certain conditions, both inflammable and explosive. At each entrance to the fuel-preparing building a permanent sign reading as follows should be conspicuously placed:

"INFLAMMABLE—KEEP LIGHTS AND FIRE AWAY—SMOKING PROHIBITED—AVOID FUEL-DUST LEAKS."

3. A leakage of pulverised fuel is to be avoided the same as a leak of steam, gas, or fuel oil.

4. All bins, elevator casings, conveyor boxes, dryers, and pulverizers should be kept absolutely dusttight.

5. Wood or other combustible material must not be used under any circumstances in connection with bins or containers for wet, dry or pulverized fuel.

6. Accumulation of pulverized fuel on surfaces where it may be dislodged and become suspended in the air, or the formation of any dust cloud, is to be avoided, and the building and equipment for preparing the fuel should be kept clean at all times.

7. Fuel dust should be swept off the machinery and other surfaces, and not blown off with compressed air.

8. Only incandescent lamps equipped with heavy guards should be used for lighting in fuel-preparing plants. Where it is necessary to use a light in a confined space, such as an elevator casing, separator or bin, this should be done with great care, so as not to allow the lamp to be broken, and the bulb should never be allowed to lie against pulverized fuel in storage.

9. The use of matches, open fires, lanterns ,or torches about the pulverizing plant should not be allowed.

10. Smoking about the premises should be forbidden.

11. The temperature of the gases at the discharge end of the fuel dryer should not be allowed to exceed 400 degrees as registered by the pyrometer placed in the flue to the exhauster or stacks.

12. In starting fire in the dryer furnace use the smallest amount of waste or wood possible. Under no circumstances should shavings be used.

13. The use of air for the transfer of pulverized fuel from one container to another should be avoided.

14. In shutting down the plant at the end of a day's work all fuel should be fed from the dryer, the dried coal bins, and the pulverizer.

15. The National Board of Fire Underwriters* should be requested to pass upon any terminal arrangement put into effect for the handling of pulverized fuel.

SUPPLEMENTAL INSTRUCTIONS.

16. Local conditions may necessitate the establishing of additional instructions.

EMERGENCIES.

17. Should pulverized fuel in bulk ignite when in storage bins or containers other than those directly connected with the furnace combustion chamber, the containers should be flooded with water through inlets, letting this water with the coal pass through the outlets.

* Insurance authority.

18. Should pulverized fuel in containers connected with the furnace combustion chamber become ignited, it should be fed to the furnace in the regular manner as quickly as possible.

INSTRUCTIONS FOR FUEL FEEDING, BURNING, AND FURNACE EQUIPMENT.

1. Be sure the stack damper is open before the fuel feed is started.
2. Be sure feeder air is turned on full before the fuel feed is started.
3. Be sure fuel feed is entirely stopped before feeder air is shut off.
4. Fuel as delivered to pulverized fuel bins should have less than 1 per cent of moisture, and should be pulverized so that 82 per cent will pass through a 200-mesh sieve.
5. All bearings should be kept well lubricated, a good dynamo oil being used on motors and blower bearings and a good grade of cup grease on the other bearings.
6. Before firing the boiler be sure to open the stack damper to give about 1/10-inch draft in the furnace. Hang ignition fire of oily waste about 12 inches from the mouth of the burner, or build a wood fire on the floor of the mixing oven, according to the nature of the fuel that is being used.
7. Be sure that the feeder blower is up to speed, and that air at 6-inch water pressure is passing through the feeder and burner.
8. After making sure that the ignition fire is burning and full air blast is passing through the burner nozzle, start one feeder at about 25 revolutions per minute. Fuel should ignite at once. As soon as the fire begins to burn well, start the second feeder—this applies to mixing ovens having twin burners—at about the same speed as above.
9. Speed up the feeders gradually and adjust the auxiliary air openings and stack damper until the boiler is under the required load. Where two feeders are used their speeds should be about equal.
10. While the boilers are under load the only attention required is to maintain the water at correct level, keep the furnace free of ash, and clean the burner mouth and auxiliary air openings when necessary.
11. Blow off heating surfaces every six hours to maintain high efficiencies.
12. To put out the fire or bank the boiler, stop the fuel feeders completely, shut off air from the feeders and close the stack damper and auxiliary air openings. This will keep the boiler hot and protect the brickwork.

The Holbeck system of handling pulverized coal fuel differs from most other systems in that the pulverizing and storage facilities are at some distance from the factory or works containing the furnaces which are to be served. The Bonnot Company, of Canton, United States, which manufactures the apparatus for this system, has no general set of rules, but issues the following advice in regard to storage:

1. Moisture in the coal should be reduced to about 1 per cent before pulverizing.
2. Pulverized coal should not be stored in bins for periods to exceed three days.
3. Pulverized coal has a tendency to collect or absorb moisture, and if allowed to stand too long spontaneous combustion will result.
4. In case spontaneous combusion does not result, the coal will bridge or arch in the bin, thus making it difficult to feed it out for use.

In this Holbeck system the coal pulverized at the distant plant, as noted above, is delivered to the furnaces by means of a distribution main. The coal is fed from the storage bins to the main plant by means of screw conveyors discharging into the suction side of a blower. It is then conveyed to the furnaces in a non-explosive mixture of air, the ratio being about 60 cubic feet of air to 1 pound of the pulverized coal. The coal which is not used at the furnaces is then returned to the bin at the main storage plant to be used over again.

Under this system, instead of having separate bins for the several

furnaces, the pulverized coal stock is available for any furnace. There is also the comparative safety of having the storage bins at the pulverizing plant and away from the high temperature near the furnaces. Furthermore, there is said to be a minimum of danger of spontaneous combustion, owing to the fact that there is continuous movement of the pulverized coal from the top to the bottom of the bins, since the coal is fed out from the bottom while that not used at the furnace is returned and enters at the top of the bin.

The following sets of rules are issued by the Fuller Lehigh Company, of Fullerton, United States:

Pulverized coal is harmless unless mixed with air; therefore the following rules and regulations should be observed:

1. No smoking, open lights, torches or matches should be permitted around a coal pulverizing plant. Use a flash light or cage-protected incandescent light. The electric-light bulb should never come in contact with the fuel in storage.

2. Never attempt to enter any tank or bin that contains pulverized coal for any reason whatever. Never insert your head within for investigation without first thoroughly airing the interior. Pulverized coal-storage tanks may contain methane, an odorless gas, which will quickly render unconscious anyone breathing it. When necessary, tanks may be aired by opening the manhole doors. When a smolder or fire exists in tanks or bins they should be closed tight. This will tend to smother the fire due to lack of air to support combustion.

3. The pulverizing mill building should be well ventilated.

4. Storage bins, containers, or conduits must be kept airtight and free from pockets and leaks. Corners or projections should be avoided.

5. Surfaces where pulverized coal accumulates should be kept clean.

6. Do not open elevator casings, conveyor covers, or pulverizing mills while machinery is in operation. After shutting down the machinery allow sufficient time for coal in suspension to settle.

7. Do not hammer or make repairs on elevator casings or any other part of building while plant is in operation.

8. Do not use waste to stop dust leaks.

9. Keep all elevator casings and conveyor troughs tight, and keep the entire building as dustless as possible.

10. Make daily inspections.

11. When the plant starts, run sufficient non-combustible material through the equipment to fill all spaces where coal might lodge. Do this occasionally in order to keep such spaces filled.

12. Always keep bottom of gear casing and pulleys or belt guards clean, or, if possible, construct them in such manner that no oil refuse can accumulate or catch fire from friction or moving parts.

13. Be particular concerning cleanliness.

14. Safety devices should be properly installed for protection from accidents around all machinery.

15. When starting up a furnace using pulverized coal as a fuel, open the damper, build the fire, turn on the air and then the coal. When shutting down turn off the coal and then the air. Failure to do this may lead to a puff or back-fire.

16. Never attempt to ignite the mixture of air and coal in the furnace by reaching in or opening the doors.—"The Engineer."

ABRASIVES FOR SPECIFIC USES.

By R. G. Williams.

The manufactured emery wheel was a decided advantage over the natural sandstone, but as manufacturing conditions grew more exacting, an attempt was made to produce a wheel which would more nearly meet

the requirements. Corundum belongs to the same family as emery, but is purer. This material was more satisfactory than emery, but because the quantity was limited and because it was difficult to obtain succeeding shipments of uniform quality, even corundum was not entirely satisfactory.

The electric furnace has made it possible now to use abrasives which are constant in quality and practically unlimited in quantity. Two distinct classes of electric-furnace abrasives are in existence, one known as the aluminous group, containing aluminum oxide as the base, while the other is known as the carbide-of-silicon group, and is marketed under certain trade names as crystolon, carborundum, xolon, etc. These electric-furnace products are known as artificial abrasives.

There is hardly an article of commerce which does not come in contact with abrasives in at least some stage of its manufacture.

The following list is comprehensive enough to give a conception of the forms and classes into which abrasives can be divided:

1. Grindstones, pulpstones, and millstones.
2. Infusorial earth, Tripoli, rottenstone, pumice, etc.
3. Garnet, quartz, chilled steel.
4. Hones, oilstones, and whetstones.
5. Diamond dust and bort.
6. Emery and corundum.
7. Artificial abrasives of two classes—namely, aluminous and carbide of silicon.

Sandstone wears more rapidly on any grinding operation than would a wheel containing an artificial abrasive, but since sandstone costs much less than a modern grinding wheel, it requires consideration of wear and cost to determine which is the more economical.

Artificial abrasives in themselves have no effect upon the human system, in that the various human fluids have no chemical or physiological action on the abrasive. However, no doctor would approve a man working in an atmosphere containing a large amount of abrasive dust. The irritation which would result in the nose, throat and lungs would in turn produce a situation whereby such diseases as tuberculosis might develop. The dust from all dry-grinding operations should be removed. Just why sandstone should cause lung diseases to develop rapidly has not been determined. Nevertheless, it is a fact that even uneducated operators are forced to give up work which requires the use of sandstone, after a year to a year and a half.

Infusorial earth, Tripoli, rottenstone and pumice are comparatively soft abrasives, and readily break down into the form of a fine powder, so that when used with water a mud is formed which aids in polishing.

Garnet is also a natural mineral, very much softer than either emery or corundum. On account of its chemical composition it cannot be bonded in the form of a vitrified wheel, but must be used in the form of layers or strips held to a background by means of glue or cement.

The nature of garnet makes it particularly well adapted for working on wood, and garnet discs find extensive use in the pattern shop for this reason.

Quartz is used mostly for grinding glass, and ordinary sand of good quality is the form in which it finds commercial use.

The action of abrasives when used in the form of hones, oilstones and whetstones differs from that when used in the form of wheels in that the cutting action is slow. As a very keen, sharp edge is essential to the metal being cut, it is therefore necessary that the abrasive be of small size. The stones are used either with water or oil, or, in rare instances, dry. Stones containing artificial abrasives have largely replaced natural stones.

In the form of dust, diamond and bort are used for lapping, or for cutting and polishing gems. A lap consists of soft metal, such as copper

or cast-iron, in which is imbedded small particles of the diamond dust or bort. The lap is revolved, rotated, or reciprocated at slow speed, and is used only where very high refinement is required.

Emery is a natural mineral. At present the mines producing the greatest tonnage are in Turkey and Greece. It is a mechanical mixture of crystalline aluminum oxide and magnetic oxide of iron, the ratio being, roughly, 60 to 65 per cent aluminum oxide and 40 to 35 per cent magnetic oxide of iron. Since magnetic oxide of iron has very low abrasive properties, emery is not an efficient abrasive, in that only approximately two-thirds of it is of such a nature as to cut effectively. Due to the fact, however, that it does contain so much magnetic oxide of iron, it bonds very satisfactorily, and becomes extremely hard in the form of vitrified grinding wheels.

Corundum is a purified form of emery, in that it is 80 to 85 per cent crystalline aluminum oxide. A very good grade of corundum was discovered in Canada, and placed on the market shortly after 1900. The supply is limited, and as the artificial abrasives have come into use, Canadian corundum has diminished. There is a good grade of corundum in India, but not an important factor in the American market.

There are various artificial aluminous abrasives on the market, all differing somewhat in both physical properties and chemical composition. The common trade names are alundum, aloxite, boro carbon, adamite, etc. They consist essentially of 90 per cent or more of aluminum oxide, the remaining materials being known as impurities, and consisting of iron oxide, titanium oxide, and silica. Abrasives of this class are produced in the electric arc furnace. Bauxite is the most satisfactory raw material to use, although emery is also used. The manufacture consists in fusing a material high in aluminum oxide under the high temperature of the electric arc, with a reduction of the impurities. The abrasive floats on top as slag, while the impurities in a very low state of oxidation settle to the bottom, and are later separated by mechanical means and a magnetic separator after the furnace operation is completed. After the current has been shut off from the furnace the pigs are allowed to cool for a number of days before being broken up into small pieces. These small pieces are further reduced in size by powerful crushers and rolls into sizes varying all the way from pieces approximately 1/10 inch across, to particles so fine that they will stay suspended in water for a long time.

All carbide-of-silicon abrasives are artificial, and also depend upon the electric furnace for their manufacture. The raw materials are coke, sand, sawdust and a little salt. The resistance furnace is used, and a cross-sectional view would show a central core of granular carbon, around which is placed a mixture of the coke, sand, sawdust and salt. The heat caused by an electric current passing through the carbon core is sufficient to result in a union of the carbon in the coke and the silica in the sand, producing a compound containing one atom each of carbon and silicon.

Pure carbide of silicon is probably colorless. The purest form known is light green, while the commercial varieties vary from green to a product dark blue to jet black in color. The chemical composition of the different kinds varies but slightly, but this difference, together with differences in furnace treatment, produces abrasives which vary enough in their physical properties to affect their commercial uses.

Numbers are used to designate the different sizes of abrasives. Thus a No. 24 abrasive means that that particular size of grain has passed through a 24-mesh screen, but is too large to pass through the next finer screen, which has 30 holes to the linear inch. Sizes finer than 200 are designated by letters, and are referred to as flours, because the particles are so fine that it is hardly accurate to refer to them as grain.

Coarse grains are used for operations where the rapid removal of stock is necessary, while the finer sizes are used in wheels intended to remove

a small amount of material and produce a good finish on the work. The flours are used in very fine oilstones, rubbing bricks, etc.

Mention should be made of the use of abrasives for polishing purposes. For this work the abrasive is not bonded in the form of a solid wheel, but is held to suitable wheels of canvas, leather, etc., by means of glue.

The object in polishing is not to remove material rapidly, but rather to take small shallow cuts, so that eventually the surface presents an attractive appearance. Aluminous abrasives are used mostly for polishing. If carbide of silicon could be readily held to a polishing wheel by means of glue, this abrasive would be used more extensively than it is.

Hardness is undoubtedly the most influential property of an abrasive. Mineralogists use what is known as Mohs' scale of hardness, in which the diamond receives a value of 10, corundum is numbered at 8, while topaz is numbered at 8. This scale is undoubtedly misleading, in that there is a much greater difference in hardness between diamond and corundum than corundum and topaz. Nevertheless it is in common use, principally because materials harder than corundum present great difficulty in being properly tested for hardness. Tests have been made indicating that the artificial aluminous abrasives are considerably harder than corundum, though not nearly so hard as diamond, while carbide of silicon is between the diamond and the artificial aluminous abrasive in hardness.

Next to hardness the most important physical property of an abrasive is toughness. Toughness is very desirable. While the individual cuts of each abrasive particle are small, nevertheless they are deep enough so that considerable resistance is brought on the grain, which, to be efficient, particularly in cutting materials of high tensile strength, must possess the property of toughness.

The reason the two types of artificial abrasives, aluminous and carbide of silicon, are in such extensive use, is primarily because carbide of silicon, while extremely hard, is not so tough as the aluminous abrasives, which in turn possess less hardness than carbide of silicon, but are much tougher. Practical experience extending over a period of years has proven that for materials of high tensile strength the aluminous abrasives are best adapted, while for materials of low tensile strength carbide of silicon is most efficient. The meeting point is probably at malleable iron. For materials of less tensile strength than malleable iron, such as brass and bronze, granite and marble, and leather and wood, it will be found that carbide of silicon will produce more satisfactory results than will an aluminous abrasive; while for steels in general an aluminous abrasive should be used for satisfactory results.—"Industrial Management."

FLAKES IN ALLOY STEELS.

The occurrence of "flakes" in "tender" nickel-steels and nickel-chrome-steels, leading to the rejection of forgings of these alloys, was one of the unpleasant experiences made during the war, especially in the United States. Cracks would develop suddenly, without apparent reason, in such steels when under test, and could not always be cured by heat treatment. The American Institute of Mining Engineers discussed the question recently in connection with papers by C. Y. Clayton, F. B. Foley, and F. B. Laney, and by H. S. Rawdon, and H. M. Howe presented a report on behalf of the Steel Ingot Committee of the National Research Committee. These papers were noticed in *Metallurgical and Chemical Engineering* of March 1 last, and a further instructive communication on the subject was contributed to the March 15 issue of the same journal by Frederico Giolotti, the Italian metallurgist, who was in New York at the time. The flakes, also known as snow-flakes, scabs, goose eggs, silver streaks, commonly appear as bright streaks, sometimes greenish in the

central portion, in alloy steels which show weakness when worked. The defect is brought out in the macrostructure by etching; radiographs show white lines; micrographs indicate lack of grain refinement. The flakes or streaks are supposed to be cracks in the steel itself; they are not found in steel which welds readily, but are troublesome in nickel-steels which do not weld. Clayton and his collaborators consider that there is always evidence of overheating in flaky steel, and they adduce a good deal of experimental evidence favoring his view; the flakes, they say, pass through the ferrite existing between sorbitic grains, and also through the grains themselves. Rawdon believes that the flakes represent original crystal faces which formed during the solidification of the inhomogeneous alloy steel; the rupture would take place already in the ingot owing to the different expansion coefficients of the metals and alloys. Howe inclines to the latter view. Flakes, he points out, also appear in alloy steel which has not been heated above 900 degrees C., and are not characteristic of the zones which were hottest in the ingot. They vanish in some steels, as the nickel percentage is reduced, he admits, whilst they increase in alloy steel which has been killed in the furnace. Tender steel should hence be humored and so cast that excessive cooling be avoided, using round molds, stripping as soon as the walls are strong and cooling by burying in ashes. This treatment would tend to favor dendritic structure, but it would lessen the flaking. In their *résumé* Howe and Rawdon agree that flakes originate in the ingots of tender alloys as intercrystalline shrinkage cracks and are probably intensified by inclusions of segregations. Giolotti would accept this conclusion; he would say that the flakes originate in tender alloys as intercrystalline cracks, but he does not believe that they are formed in the ingot. He remarks that the Ansaldo Company, with which he is connected, made 10,000 guns for the Italian Army, and had no rejection on account of flakes, though they were troubled with them. His chief point is: to prevent flaking, keep not only sulphur and phosphorus as low as possible, but avoid all iron oxide, do not use rusty turnings or borings, and fuse in a reducing atmosphere under a thin covering of silica slag. He adds the ferrochrome 20 minutes before pouring (hardly any chrome should in good practice go into the slag) and pours hot and cools in the air—contrary to Howe; he has not observed any preferential directions of the flakes in molds of various shapes (Howe). He ascribes the trouble to the diffusion of emulsified, microscopically undetectable iron oxide through the metal. That non-metallic inclusions are surrounded by decarbonized zones he explains by the presence of both CO and CO_2 in the steel (dissolved, not in cavities); the equilibrium: $3Fe + 2CO \rightleftarrows Fe_3C + CO_2$ is shifted to the left by an oxidizing slag so that a zone of decarbonized iron is produced, supersaturated with CO. There is no internal crack, however, until the material is strained above the elastic limit.—"Engineering."

THE TONNAGE OF MODERN STEAMSHIPS.*

By A. T. Wall, A. R. C. Sc., Member.

1. The main object of this paper is to focus attention on the effect of recent legislation and modern machinery on tonnage measurement. Some remarks on the basis of measurement are also made.

2. *Basis of Measurement.*—It will not be out of place to touch on the principle underlying the whole question, especially as it is thought by many that the present method is not only complicated but unfair as between ship and ship. The subject is obviously quite unsuitable for exact mathematical

* Paper read before the Institution of Naval Architects, April 11, 1919.

treatment, and it must be recognized that whatever basis is used, anomalies are bound to occur.

3. This side of the matter has been dealt with by Mr. E. R. Johnson in his book on "Measurement of Vessels for the Panama Canal." He gives several bases that have been proposed for tonnage measurement, and considers them in a general way in coming to the conclusion that the only fair and practicable method is the present one of assessing the earning spaces in the ship. The measurement of vessels for the Panama Canal is very similar to that under the Suez Canal Rules, which is the same in principle in all maritime countries.

4. The proposed bases are as follows:

(1) Gross deadweight.
(2) Cargo deadweight.
(3) Block displacement (length × beam × draught).
(4) Displacement.

5. Gross deadweight is obviously an unfair method, as it takes no account of the passengers carried in the ship usually having a small deadweight.

6. Cargo deadweight would be very difficult if not impracticable to assess, and, to be logical, would vary under different conditions, requiring repeated measurement. The same objection as to passengers applies as in paragraph 5. Further, ships carrying no cargo would have no tonnage.

7. Displacement tonnage would appear to be fairer than the foregoing. This tonnage is made up of the light weight of the ship and the gross deadweight. The former is heavier in a passenger ship than in a cargo vessel of the same length for many reasons; among these are greater superstructure and heavier machinery. In addition, large quantities of fresh water, stores, baggage, &c., must be carried for the passengers. This excess weight gives a measure of the passengers carried.

8. A great advantage of displacement tonnage would be that it is easily measured, and a certified displacement scale would be carried by the ship. The tonnage on this basis would also vary with the loading of the ship and so be more commensurate with her earning capacity and the services rendered to her for which dues are levied.

9. Block displacement tonnage is similar to displacement tonnage, but is severe on vessels of fine form.

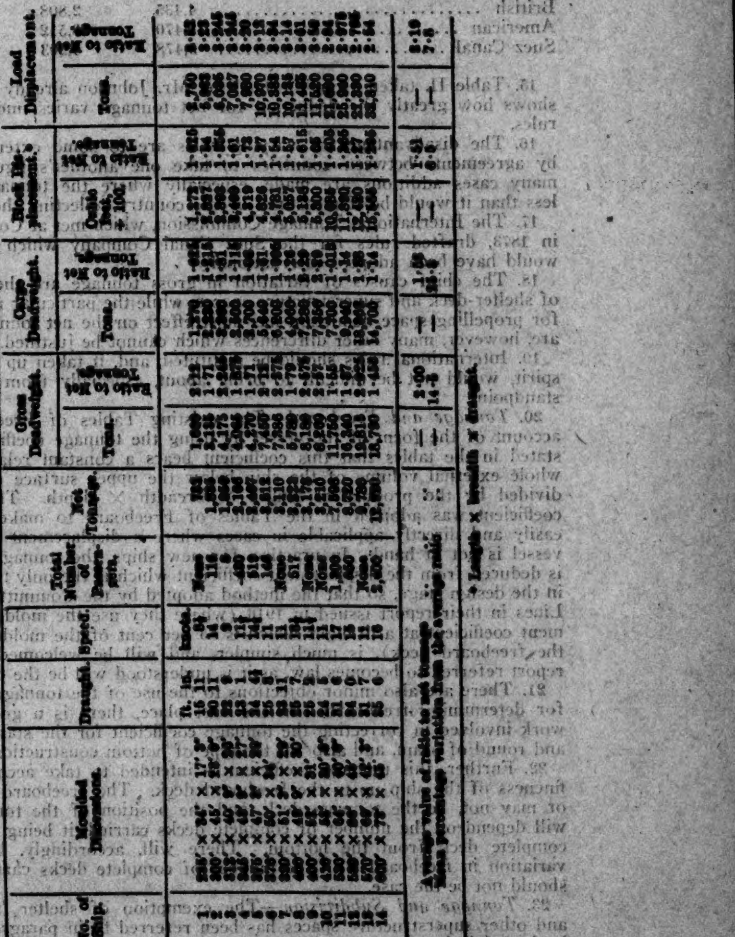
10. It would be impossible by any system to take an accurate measure of the value of the cargo and the revenue from passengers, even if this were a proper basis for tonnage dues, which is doubtful. It would be equally impossible to measure properly the actual service performed by any port, canal, &c., to a particular ship. Any basis must be somewhat arbitrary, but the existing method does, or at least could, be made to give a true measure of the earning spaces in a ship.

11. Table I gives particulars of various vessels, showing how the bases compare.

12. From Table I it is evident that no two bases will give corresponding results, but it shows that there might be some justification for a displacement basis or a block displacement basis.

13. *International Tonnage.*—Although the principle of measuring the actual earning spaces of a ship is adopted in all maritime countries, there are many variations in details which give the same ship very different tonnages under different rules. Gross tonnages do not vary greatly except on account of the exemption for shelter-deck and certain superstructure spaces as allowed under British rules. This causes very different values, for instance, between American and British measurements. One example is sufficient as an illustration. A British ship, with shelter-deck spaces

exempt, has a gross tonnage of 4,435, whereas under American rules it is 5,470, an increase of 23 per cent. The German rules make the same allowance as the British. It is often maintained that there is no justification



for the exemption of the shelter 'tween-deck spaces. The tonnage laws of the American, Suez and Panama Canal authorities do not recognize it.

14. The allowances for net tonnage are very varied. Taking the same ship as in paragraph 13, her tonnages are as follows:

	Gross.	Net.	Percentage Increase Net Tonnage over British.
British	4,435	2,808	—
American	5,470	3,512	25
Suez Canal	5,478	4,203	50

15. Table II, taken from the book by Mr. Johnson already referred to, shows how greatly the deduction for net tonnage varies under different rules.

16. The disadvantages of varying rules are to some extent overcome by agreements between countries to take one another's figures, but in many cases additions are made, especially where the tonnage is much less than it would be if measured by the country collecting the dues.

17. The International Tonnage Commission, which met at Constantinople in 1873, drafted rules for the Suez Canal Company which it expected would have been adopted internationally.

18. The chief causes of variation in gross tonnage are the exemption of shelter-deck and superstructure spaces, while the particular rule adopted for propelling space deduction has most effect on the net tonnage. There are, however, many other differences which cannot be justified.

19. International rules should be instituted, and, if taken up in the right spirit, would not be difficult to bring about, especially from a technical standpoint.

20. *Tonnage and Freeboard.*—The existing Tables of Freeboard take account of the form of the vessel by using the tonnage coefficient. It is stated in the tables that this coefficient bears a constant relation to the whole external volume of the ship below the upper surface of the deck divided by the product of length × breadth × depth. This tonnage coefficient was adopted in the Tables of Freeboard to make the tables easily and directly applicable in cases where a displacement scale for a vessel is not at hand. In practice, for new ships, the tonnage coefficient is deduced from the displacement coefficient which is the only thing known in the design stage, so that the method adopted by the Committee on Load Lines in their report issued in 1916 (where they use the molded displacement coefficient at a draught which is 85 per cent of the molded depth to the freeboard deck), is much simpler, and will be welcomed when the report referred to becomes law, as it is understood will be the case.

21. There are also minor objections to the use of the tonnage coefficient for determining freeboard. In the first place, there is a good deal of work involved in correcting the tonnage coefficient for the standard sheer and round of beam, and also for the type of bottom construction adopted.

22. Further, this tonnage coefficient is intended to take account of the fineness of the ship up to the freeboard deck. The freeboard deck may, or may not, be the tonnage deck, and the position of the tonnage deck will depend on the number of complete decks carried, it being the second complete deck from the bottom. There will, accordingly, be a small variation in freeboard with the number of complete decks carried, which should not be the case.

23. *Tonnage and Subdivision.*—The exemption of shelter 'tween-deck and other superstructure spaces has been referred to in paragraph 13. It is doubtful, in passenger ships, owing to the present subdivision rules, whether it would be possible to exempt many such spaces.

24. To be exempt from tonnage, a shelter 'tween-deck must have a tonnage opening, openings in the transverse bulkheads, and scuppers discharging overboard from the deck under the shelter deck, that is, the upper deck.

25. The freeboard of such vessels brings the load draught only a little below the upper deck. If the upper deck is retained as the bulkhead deck it generally becomes necessary to put in so many bulkheads to meet the subdivision requirements as to make the ship impracticable. The alternative is to carry the watertight bulkheads up to the shelter deck. The necessary tonnage openings in these bulkheads cannot then be fitted, and the scuppers in the upper deck must also be omitted. This matter was discussed in detail by the author in a paper read before this Institution in 1916 on "Some Effects of the Bulkhead Committee's Rules in Practice."

26. The inclusion of the shelter 'tween-decks in the tonnage will increase the gross and the net tonnages. If the propelling space is still large enough with the light and air spaces to be over 13 per cent of the new gross, the increase in net tonnage will be of the order of 25 per cent. It may be, however, that the propelling space may not be large enough to be over 13 per cent of the gross, when the increase in net tonnage may be as much as 50 per cent. This is a very serious disadvantage of the subdivision rules.

27. One other case of a similar nature may be briefly mentioned. It occurs mainly with home trade passenger and cargo steamers of the three-island type. To subdivide adequately such vessels it will be necessary to fill in one or both of the wells, making the vessel flush deck forward or throughout. This adds to the tonnage, and although the wells are used to carry cargo on occasions, they are not always included in the tonnage measurement. An example of this is given in the paper referred to in paragraph 25, and the increase of net tonnage may be about 20 per cent.

28. *Deduction for Propelling Space.*—The British law, for propelling space allowance in screw steamers provides that when the tonnage of the space solely occupied by and necessary for the proper working of the boilers and machinery is above 13 per cent and under 20 per cent of the gross tonnage, the deduction shall be 32 one-hundredths of the gross tonnage. If the space does not lie between these limits the deduction is 1¼ times the actual space.

29. This rule was adopted in 1854, when tank boilers and reciprocating engines were the principal power units and coal the main fuel. Since then, and especially during the last ten years, great advances have been made with propelling machinery. Modern machinery, such as geared turbines, water-tube boilers, oil engines, electric transmission and oil-fired boilers, occupies much less space for the same power than the old machinery. Consequently modern machinery often occupies less than 13 per cent of the gross tonnage.

30. The result is a considerable increase in net tonnage, quite out of proportion with the increase in cargo space, as evidenced by the following example:

Ship with Old Machinery.

Ship dimensions (molded)	580 ft. by 70 ft. by 45 ft.	
Gross tonnage		20,000
Deductions for crew spaces, &c.	1,350	
Propelling space with coal-fired cylindrical boilers and Q.E. reciprocating engines	2,800 (14 per cent of the gross)	
Deduction on account of propelling space (32 per cent gross)	6,400	
Total deductions		7,750
Net tonnage with old machinery		12,250

TABLE II.—*Gross and Net Tonnages of the Metal Steam Vessels of the World, 1910, and the Percentage Deducted from the Gross in Ascertaining the Net.*

Flag.	Gross.	Net.	Per Cent. Deducted.
Argentinian	139,846	92,419	40
Austro-Hungarian	727,360	485,675	37
Belgian	296,912	194,336	34
Brazilian	236,425	145,582	38
British	17,040,962	10,852,899	30
Chilean	114,067	72,605	36
Chinese	86,320	54,598	35
Cuban	53,494	32,105	40
Danish	563,536	331,763	41
Dutch	985,704	607,248	38
French	1,445,432	895,018	38
German	3,959,147	2,416,570	39
Greek	486,281	316,838	35
Haitian	3,937	2,017	49
Italian	955,716	567,640	41
Japanese	1,064,169	675,983	36
Mexican	27,924	16,513	41
Norwegian	1,885,681	858,220	37
Peruvian	10,371	5,954	43
Portuguese	78,889	46,677	41
Roumanian	31,068	18,660	40
Russian	687,231	400,761	41
Sarawak	3,965	2,380	40
Siamese	12,607	7,782	38
Spanish	746,047	456,198	39
Swedish	735,909	448,873	40
Turkish	110,770	68,758	37
United States (excluding Northern Lakes)	1,689,911	999,505	34
Uruguayan	48,337	30,115	37
Venezuelan	2,166	1,256	41
Other flags	29,857	21,809	26
Philippine	33,287	20,634	38
Total (national)	34,039,927	21,115,066	39
Suez Canal	23,054,901	16,581,898	28·07

TABLE III.

Type of Ship.	Type of Machinery.	Percentage Reduction in Propelling Space on Space Occupied by Old Machinery.
1. 600-ft. intermediate passenger and cargo steamer	Coal-fired cylindrical boilers and double reduction geared turbines	13
2. ,, ,,	Oil-fired water-tube boilers and double reduction geared turbines	35
3. 530-ft. intermediate passenger and cargo steamer	Coal-fired cylindrical boilers and double reduction geared turbines	10
4. 400-ft. cargo steamer	Camellaird-Fullager oil engines	37
5. 400-ft. cargo steamer	Coal-fired cylindrical boilers and Ljungstrom turbo-electric engines	10
6. 150-ft. coaster	Camellaird-Fullager oil engines	20

Ship with Modern Machinery.

Gross tonnage (as before)		20,000
Deduction for crew spaces, &c. (say, as before)		1,350
Propelling space with oil-fired water-tube boilers and double-reduction geared turbines (10 per cent. of gross)		
Deduction on account of propelling space, 10 per cent. by 1¾ per cent of gross	3,500	
Total deductions		4,850
Net tonnage with modern machinery		15,150
Increase in net tonnage		24 per cent.
Increase in cargo space		5 per cent.

31. The propelling space with the old type of machinery may be so little above 13 per cent of the gross tonnage that the substitution of geared turbines of the same power for reciprocating engines, and retaining the coal-fired cylindrical boiler, decreases the engine-room space and the boiler-room space so much as to bring the propelling space below 13 per cent of the gross tonnage. Actual cases of this have come under notice.

32. Some actual cases of the percentage reduction in propelling spaces are given in Table III.

33. The progressive shipowner who adopts modern machinery will therefore generally lose money by paying high tonnage dues, abroad as well as at home, or by deliberately making unnecessarily large machinery and light and air spaces and so wasting cargo and passenger space. The latter course may not be successful, as the Board of Trade have power to refuse to allow space in the measurement if it is not reasonable in extent for the intended purpose.

34. There is, of course, some gain in cargo space to set off against the increase in net tonnage, but it is not proportional. Examination shows that the result is a net loss. In case 2, Table III, the gain in cargo space is 6 per cent and increase in net tonnage 26 per cent. The annual loss in revenue has been estimated to be 4,000*l.*

35. The present rule fixed 13 per cent to provide comfort for those who had to look after the machinery, and the 32 per cent reduction was made to compensate for bunkers. There does not appear to be any justification for this percentage rule, and in view of its unfair incidence in many modern vessels, the question naturally arises as to whether it ought not to be dropped, and replaced by a rule which gives some multiple of the propelling space as allowance for bunkers.

36. To say that, as yet, few ships are affected is not strictly true, and even if it were so legislation should anticipate progress and help it.

37. As the British tonnage regulations have been adopted by a large number of nations, alterations would probably have to be made by international agreement, and the British law now regulating tonnage measurement would have to be altered by Act of Parliament. In the meantime a proposal is put forward for meeting the case. It is not thought fair that ships with propelling spaces not exceeding 13 per cent of the gross tonnage should have the 32 per cent deduction. To make matters more equitable it would appear reasonable to make the deduction the same percentage of 32 per cent as the actual propelling space is of 13 per cent. For example, a ship with 10 per cent space would have a deduction of $10/13 \times 32$, that is 24.6 per cent instead of 17.5 per cent under the present rules. Under this proposed rule the ship in case 2, Table III, would then lose practically nothing as compared with the same vessel fitted with the old type of machinery.—"Engineering."

STANDARDIZATION OF MARINE BOILERS.

The movement first initiated on the North-East Coast towards the standardization of marine engines, boilers and accessories led ultimately to the formation of a National Committee, to which was delegated the duty of considering the question in all its bearings. The committee, which is composed of members appointed by the Institution of Naval Architects, North-East Coast Institution of Engineers and Shipbuilders, Institution of Shipbuilders and Engineers in Scotland, Liverpool Engineering Society, Institute of Marine Engineers, and on which Lloyd's Register, the British Corporation, the Bureau Veritas, the Board of Trade, the Barrow and Belfast districts, as well as steel manufacturers, are represented, has been at work for some time past. Mr. A. E. Seaton has been chairman, and Mr. J. T. Milton has acted as vice-chairman.

The first subject which it was decided to investigate is that of marine boilers and accessories, and a report has now been prepared which deals with main boilers of the cylindrical type, auxiliary and special boilers, and with steam feed and other pipes subject to internal pressure. The opinion is expressed that at present it would serve no useful end to attempt a standardization of boiler design. The committee thinks, however, that standardization should be at once carried out in respect of the various component parts, whereby economy in their manufacture might be affected without placing restriction on general design. Special consideration has been given to the practice of the British Admiralty and the Naval and Mercantile Authorities which control boiler construction in foriegn countries. The committee has gathered a considerable amount of valuable evidence respecting the behavior of boilers of the type reported upon when working under service conditions, and has ascertained which parts are most liable to give rise to trouble. It has had some experiments carried out with a view to elucidating some points on which sufficient evidence was not forthcoming, and is having further experiments made with the same object, so that the evidence on which conclusions are based may be convincing.

The committee is convinced that basic open-hearth steel as now produced may be used with advantage in the construction of any part of a marine boiler, the tests and conditions imposed being the same as those on acid open-hearth steel. It considers that in a general way the standard 28-32 tons tensile steel should be used for the cylindrical shell plates and long stays; and, if required, steel having an ultimate tensile strength of 35 tons may be used, but beyond 35 tons it shall be a matter of arrangement with the classifying authority, the owners and builders.

The committee is of the opinion that the steel now supplied for boilers is so good and reliable, and that boiler design and methods of manufacture are so much improved, that they command much more confidence than formerly obtained. It finds also that boilers made for the Mercantile Marine in accordance with the rules of Lloyd's Register, and that the cylindrical boilers made under Admiralty rules, which permit even lighter scantlings than do Lloyd's, have proved by many years' experience to be safe and reliable. It, therefore, has come to the conclusion that the shell plates need be no thicker than required by the rules of Lloyd's Register, and so have made the formula for thickness of cylindrical plates on this hypothesis. Seeing that all boilers are now made entirely of tested material from designs made in accordance with approved rules for their safety, and under inspection during construction, there is no need to test them by hydraulic pressure to the extent that was formerly necessary.

Two of the most important points which are raised in the report, and on which the committee is at variance with the authorities, are the hydraulic test pressure, and the factor of safety for shell plates. Marine

engineers are aware that while the Admiralty hydraulic test only calls for a margin of 90 pounds over the working pressure, the Board of Trade has insisted on this test being made at double the working pressure. The variations in this respect are indeed somewhat remarkable, the French authorities requiring 85 pounds, the Germans 75 pounds and the Italians 71 pounds in excess of the working pressure for testing purposes. The differences in practice may be illustrated in another way. A North-East Coast standard boiler for a cargo vessel which is designed to work at 180 pounds pressure must pass a Board of Trade survey at 300 pounds pressure, whereas a French boiler would be tested at 265 pounds, a German-made boiler at 255 pounds and one made by an Italian firm at 251 pounds. An Admiralty transport boiler would have to undergo a test of 270 pounds. In the case of the shell plates a boiler 17 feet in diameter, having plates 1 inch thickness, would be passed by the British Admiralty for 190 pounds working pressure, by Lloyd's for only 154 pounds, and by the British Corporation for 157 pounds. It was the existence of these anomalies that led the committee to draw up a new rule for boiler-shell plates. Under this rule a boiler which is to work at 157 pounds pressure will be tested at 285 pounds hydraulic pressure, as compared with the 280 pounds pressure, which is the existing requirement of the Board of Trade and the Admiralty, although the former authority would only allow 140 pounds working pressure. The action taken by the committee is based on the belief that every useful purpose will be served if the hydraulic test to twice the working pressure is limited to boilers, the working pressure which does not exceed 100 pounds per square inch, and that for higher working pressures it will be sufficient if the test pressure is 50 pounds in excess of 1½ times the working pressure. This will mean that the stress at hydraulic test will be less on the scantlings than was the case with boilers having a factor of safety of 4.5. In view of the fact that manufacturers of plates prefer that thicknesses should be expressed in 32nds of an inch, rules involving thicknesses of plates have been based on that condition.

After consultation with tube makers the committee has drafted a specification which will be worked to by it in future. This step is necessary, as no test conditions have hitherto been laid down for the smoke tubes of boilers or even for the stay tubes.—"The Steamship."

JAMES WATT CENTENARY.

On August 25th of this year one hundred years will have elapsed since the death of James Watt, and it is proposed to commemorate the event in a fitting manner. A provisional scheme has been drawn up and will be submitted for consideration at a meeting in the Council House, Birmingham, at 2.30 P. M. on Thursday next, May 8th. This scheme provides for a celebration with a banquet and addresses on August 25th, the endowment of a James Watt Chair in Birmingham University, and the erection of a James Watt Memorial Building which would serve as a museum of Watt, Murdock and Boulton relics and a meeting place for engineers.

THE FULLERPHONE.

On April 24th Major A. C. Fuller read a paper before the Institution of Electrical Engineers in which he described the Fullerphone and its application to military and civil telegraphy.

During the early period of the war it was discovered, he explained, that the vibrator system of telegraphy, using the telephone receiver, which

system had shown itself admirably suited to military requirements in previous wars, possessed certain very serious disadvantages of which the following were some of the most important: (*a*) Such vibratory currents induce similar currents in other lines in their proximity; hence the number of possible lines on one route was strictly limited owing to the interference. (*b*) Owing again to the vibratory character of the current, the length of lines over which the vibrator can be used is strictly limited by their capacity. (*c*) By the use of suitable apparatus these vibrating signals can be read at immense distances from the line, and hence any messages can be overheard by listening sets. (*d*) The use of vibrators in any area effectually jams in that area friendly listening sets which might otherwise overhear enemy messages. (*e*) It is manifestly impossible to superpose the vibrator system on telephone lines.

Fig. 1

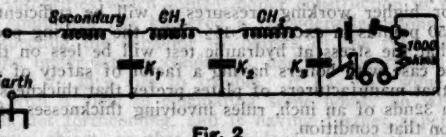


Fig. 2

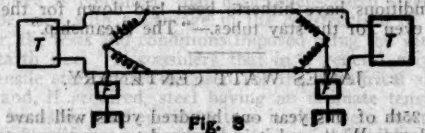


Fig. 3

In order to produce something more suitable than the vibratory system, it was necessary to retain as many of its good points, at the same time substituting something less troublesome than the vibratory current. The solution appeared to be the provision on the line of a direct current, as in ordinary single-current Morse, but of as low a value as possible—at the same time converting the energy thus transmitted into vibrating or pulsating current at the distant end—and further, such vibrating or pulsating currents had essentially to be closely confined to the distant end and prevented from surging back along the line. These ideas are embodied in the receiving and sending circuits shown in Figs. 1 and 2. K_1, K_2, K_3 are condensers, each of 1 microfarad. C H_1, C H_2 are choke coils of large impedance, each about 2·4 henries and 750 ohms resistance. The chopper is an electrically-driven vibrating interrupter fitted with two contacts, one of which is used to drive it, the other to interrupt. The

chopper or buzzer can be adjusted to any frequency between 300 and 700. Now, if a steady electromotive force is applied between the line and earth, Fig. 1, and the circuit is closed at X, a steady current will flow through the choke coils, contact 2, and the receiver. If the circuit is broken at X, the current cannot flow through the receiver, but will flow into and charge the condensers. When the circuit is again closed at X, the condensers partially discharge through the receiver R.

When the interrupter X is working we therefore get an intermittent current in the telephone receiver, giving an audible note, while, if the choke coils and condensers are suitable, the line current alternately flows through the receiver and into the condensers and remains practically constant and continuous in the line. The result is that the dots and dashes sent by the single-current Morse key at the far end are received as short or long notes in the telephone receiver at the receiving end, while the current in the line is of much the same nature as that sent by an ordinary single-current Morse set, except that it is very much smaller, as readable signals can be obtained with about ½ micro-ampère. In practice a main battery of one dry cell is used.

The arrangement of choking coils and condensers not only prevents any appreciable variation in the line current, but also prevents any vibrating currents—such as are produced by induction from other circuits, by a buzzer on the line, or from a telephone—from passing through the operator's head telephones.

When the sending battery is connected as shown in Fig. 2, the current can only rise in the line comparatively slowly, owing to the effect of the capacity of the condensers and the self-induction of the choke coils. The object of this retardation of the rise of current is to prevent clicks being heard in a telephone receiver inserted in the line. This has a twofold object: (*a*) To prevent the possibility of Morse messages being read from these clicks; (*b*) to prevent the clicks interfering with telephony carried on over the line simultaneously with Morse signalling.

It is apparent that a telephone can be added on the same line without any interference between the Fullerphone and the telephone as far as speech is concerned. Fig. 3 shows how the Fullerphone may be—and is, in fact—superposed over telephone pairs.

The interrupter buzzer is used for calling with the Fullerphone, and for this purpose its ordinary battery is reinforced by two additional cells, making three in all. This call is considerably louder than the Fullerphone signals if the line is comparatively short and in reasonable condition, but if the line is of very high resistance or capacity the Fullerphone signals will be louder than the buzzer signals. It must be remembered that the buzzing call has all the disadvantages of the buzzer, and consequently must never be used near the front line for sending any call or signal which it is important that the enemy should not overhear.

The author then proceeded to discuss the effect of earth and leakage currents on the working of the instrument. Such disturbances may be eliminated by means of a potentiometer, which is connected in series in the line and impresses a portion of the electromotive force of the cell on the line. If the electromotive force impressed on the line is equal and opposite to that producing the earth current, the earth current and its disturbances are eliminated. Fig. 4 shows the circuits of the latest pattern Army field set. This instrument provides for telegraphy and telephony either independently or simultaneously on the one line without further apparatus. When the receiving buzzer is not working the buzzing call from a distant station is heard both in the head receivers and in the hand set, or, if the hand set is not plugged in, only in the head receivers. When "Fullerphoning" the Morse signals are heard in the head receivers only. When speaking and telegraphing the speech is heard in the hand set and the Morse signals in the head receivers.

Major Fuller than described a number of tests which were carried out before the system was adopted for the Army. Fig. 5 is a reproduction of oscillograms made by the Research Department of the G. P. O. Curves A and B are to the same scale and taken under normal conditions, and show that the intermittent currents in the receiver are considerably greater in magnitude than the line current due to their short duration. The normal variation in the line current is scarcely discernable, but this is shown in curve C by using an excess line voltage.

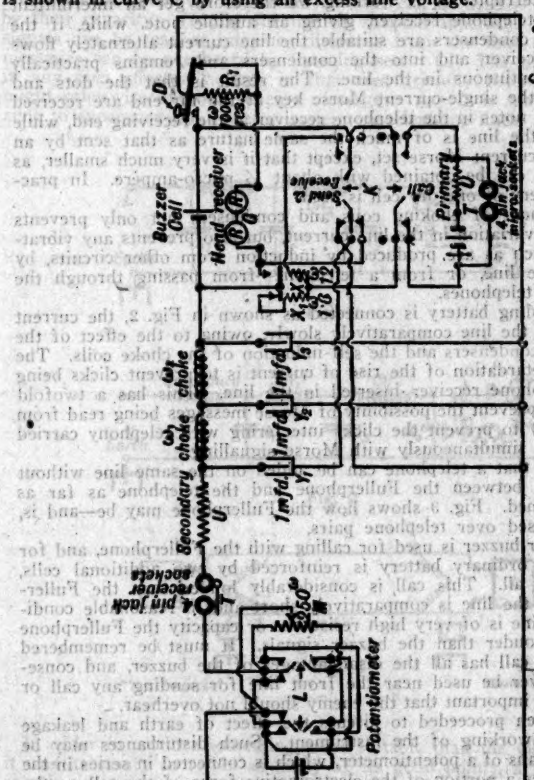
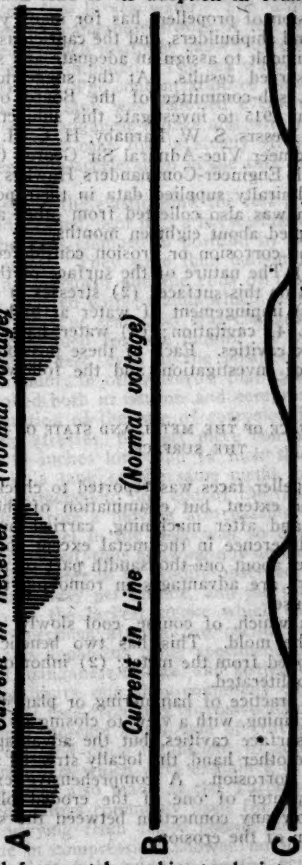


FIG. 4.—ELECTRICAL CONNECTIONS OF FULLERPHONE ARMY FIELD SET.

Major Fuller dealt with the possibilities of the instrument for civilian uses, and summoned up the present position as follows: For military work the Fullerphone has practically superseded the vibrating system formerly employed. It is now being tried for working long lines in back areas, where it will almost certainly replace all hand-worked Morse sounder sets. It appears, therefore, that as far as line telegraphy is considered,

the Fullerphone system will be used throughout, except—for the present, at any rate—for those lines normally working high-speed Wheatstone. For civilian use one cannot expect sounder sets that are established and giving satisfaction over well-constructed routes to be discarded in favor of Fullerphones; but it is expected that the advantages to be derived from the new system will lead to its adoption in countries not already thor-



SWAIN Sc.

"THE ENGINEER." FIG. 1.—OSCILLOGRAMS OF CURRENTS IN RECEIVER AND LINE CIRCUITS

oughly developed from a telegraphic standpoint. An opening should also arise in tropical countries where maintenance difficulties are considerable, and it is probably not too much to say that the system described would give satisfaction wherever difficulties from any cause render Morse sounder working precarious. For cable work the author feels that an investigation of the possibilities of a system combining Fullerphone, amplifiers, special existing relays, and recording and printing devices, would prove very profitable.—"The Engineer."

INVESTIGATIONS INTO THE CAUSES OF CORROSION OR EROSION OF PROPELLERS.*

By The Hon. Sir Charles A. Parsons, K.C.B., F.R.S., and Stanley S. Cook.

The corrosion or erosion of propellers has for many years engaged the attention of engineers and shipbuilders, and the capricious character of the action has rendered it difficult to assign an adequate and satisfactory cause to account for the observed results. At the suggestion of Professor H. C. H. Carpenter, a sub-committee of the Board of Invention and Research was formed in 1915 to investigate this subject. Those serving on this committee were Messrs. S. W. Barnaby, H. C. H. Carpenter, S. S. Cook, J. H. Gibson, Engineer Vice-Admiral Sir George Goodwin and Sir Charles A. Parsons, with Engineer-Commanders Hawkes and S. R. Dight as secretaries. The Admiralty supplied data in their possession on this subject, and information was also collected from other available sources. The investigations occupied about eighteen months.

The possible causes of corrosion or erosion considered may be stated under five headings: (1) The nature of the surface of the metal and the state of the initial stress on this surface; (2) stresses in the blades under working conditions; (3) impingement of water at high velocity on the surface of the blades; (4) cavitation; (5) water hammer produced by the closing up of vortex cavities. Each of these possible causes in turn was made the subject of investigation, and the following is a general account of the results.

(1) Nature of the Surface of the Metal and State of Initial Stress in the Surface.

The machining of propeller faces was reported to check erosion of the propeller blades to some extent, but examination of the material of a propeller blade before and after machining, carried out by Professor Carpenter, showed no difference in the metal except for a thin film of scale, &c., on the surface, about one-thousandth part of an inch in thickness. Nevertheless, there are advantages in romoving this scale before the propeller is put into use.

Large propeller blades which, of course, cool slowly, evidently anneal well while cooling in the mold. This has two beneficial results: (1) Casting stresses are removed from the metal; (2) inhomogeneities present in the metal as cast are obliterated.

One firm had made a practice of hammering or planishing the driving faces of blades after machining, with a view to closing up minute openings that might lead to sub-surface cavities, but the advantages, if any, are open to question. On the other hand, the locally stressed metal surface is possibly more liable to corrosion. A comprehensive examination was made by Professor Carpenter of one of the eroded blades of a fast cruiser. It failed to show any connection between the structure of the metal and the distribution of the erosion.

(2) Stresses in the Blades under Working Conditions.

The committee initiated special experiments to determine the action of sea water on manganese-bronze under stress. These experiments were carried out by Professor Carpenter at the Imperial College of Science and Technology.

* Institution of Naval Architects. Abridged.

For studying the stress below the elastic limit, two machined strips of the metal, 1⅛ inches thick, which were perfectly flat in the unstrained condition, were strained by clamping the ends firmly together, while at the middle they were held apart by a ⅜-inch rod of the same metal. The assemblage was suspended in a glass tank by glass rods and was immersed to about the middle line in sea water. The whole of the metal in this apparatus consisted of manganese-bronze of the same composition, so that all disturbing effects which might be caused by electrolytic action between different metals were eliminated. The sea water used was obtained from the Channel off Dover. The concentration of the water was kept constant and the charge in the tank was renewed every week. This experiment, made both with cast manganese-bronze and with rolled manganese-bronze, did not show in either case that a steady stress throughout the range given above caused any acceleration in the rate of corrosion of the metal in sea water.

For stresses beyond the elastic limit the test plates were removed from the apparatus and cleaned, bent into a horse-shoe form by steady squeezing in a vice, and the ends clamped in position by a steel nut and bolt insulated from the metal. They were then half-immersed in sea water, the entire gripping arrangement and the ends of the strip, for a distance of 1½ inches being covered with paraffin which protected them from contact with the sea water. The sea water was changed from time to time and the experiment continued for one month. It was found that no action took place in either test, even at the position where the tensile stress would be a maximum; in other words, that steady stress well above the elastic limit, applied both in tension and compression, does not appear to cause any acceleration of the rate of corrosion in sea water.

For alternating stresses, the test piece consisted of a strip of metal 3 inches wide, 10 inches long and ⅛ inch thick. This test piece was placed endwise in a tank of the same metal, one end being fixed to a stationary clamp attached to one end of the tank, the other receiving an oscillatory motion from a moving arm passing through the sides of the tank. The water level in the tank was so adjusted that only half the test piece was immersed, thus allowing any acceleration of the corrosion at the surface of the water to be readily observed. The speed of oscillation was kept at about sixty per minute, and the total stroke 15/16 inch, this being found to be the largest stroke which could be employed without fracture of the specimen after a few oscillations. In various tests carried out with this apparatus using specimens of both rolled manganese-bronze and cast manganese-bronze, and lasting over periods varying from 167 hours to 858 hours, one alloy commonly employed for propellers was found to be almost free from any corrosive action, whilst a second showed only a very slight increase in the rate of corrosion in consequence of the alternating stress, and quite insufficient to account for the condition of some of the propellers which were examined by the committee.

Experiments were also made at the Turbinia Works, Wallsend, which showed that the alloys commonly used for propellers when subjected to steady stress, varying from 13,500 pounds per square inch in tension to the same value in compression, showed no increase in corrosion due to stress when subjected, in addition, to the action of a diverging jet of sea water, in the apparatus described below—the specimen being a rod placed centrally in the diverging passage, whilst the initial pressure of the jet was sufficient to cause cavitation.

These experiments, therefore, gave valuable negative results, since the frequently observed bending of the tips of eroded blades had been thought to indicate that stress was an important factor. Calculations were made independently by Messrs. Barnaby, Gibson and Cook of the distribution of pressure and the working stresses on a propeller blade, with the con-

clusion that these stresses did not normally reach excessively high values. In the case, for instance, of the propellers of a fast cruiser, the maximum estimate of the stress was from 9,000 pounds to 10,000 pounds per square inch, and this only at the root of the blade and far removed from the place of corrosion. An estimate of the position of center of pressure furnished by the National Physical Laboratory from their experimental data of aerofoils did not indicate that any high stresses were likely to arise through irregular distribution of the pressure.

(3) IMPINGEMENT OF WATER AT HIGH VELOCITY ON THE SURFACE OF THE BLADE.

Several experiments were made with jets of water, of pressures up to 1,500 pounds per square inch, on plates of various brass alloys, including those used for propellers, with the water impinging both normally and obliquely. These produced merely an etching effect on the surface of the metal. It should be remarked, however, that the scale of these experiments was small.

(4) CAVITATION.

Numerous experiments were carried out at the Turbinia Works, Wallsend, with jets having diverging mouthpieces, supplied with sea water under pressure by a centrifugal pump, and discharging into a tank at atmospheric pressure. The velocity of the water was more than sufficient to sustain a vacuum and rupture of the water at the throat, in virtue of the diffuser action of the diverging mouthpieces. Under these conditions cavitation took place. A rod of the material to be tested, after being highly polished, was placed along the axis of the jet and there exposed to the action of the cavitating stream. The diagrams show the jets with the rods in the positions in which they were tested. The sea water was obtained from the North Sea, 3 to 4½ miles E.N.E. of the mouth of the river Tyne, at flowing tide.

In a second series of these experiments, carried out with the object of verifying the preliminary conclusions and of testing the effects under these same conditions of the additional circumstances of (1) stress, (2) the injection of the gases oxygen and CO_2 into the stream, and (3) annealing, a larger nozzle was employed, and a more powerful pump to give higher jet velocities. This nozzle was formed by two shaped slabs of brass fitted between parallel walls of glass. It was thus of rectangular cross section, 1 inch square at the throat, increasing to 1 inch by 4½ inches at the discharge. The rods were either ¼ inch diameter or ¼ inch square. The speed of the water was in all cases in the neighborhood of 90 feet per second, and under this condition cavitation occurred at the throat and extended for 5 inches or more beyond it.

In all, twenty-four different rods were submitted to this treatment. One of these was of brass, one of Monel metal, two of copper, fourteen of rolled manganese-bronze, and six of cast manganese-bronze—five of these were made from strips cut from propeller blades, in one case from a blade whose working face was deeply eroded.

The general character of the action on these rods, as revealed by the microscope, may be described as an etching of the surface, varying in degree. In the case of the Monel metal and the brass rods this action was very slight. In general, on the rods of manganese-bronze, there was incipient etching near the entry, increasing in depth towards the throat where the velocity was greatest. In some cases it continued deep to the end of the rods, and in other cases it gradually diminished from the throat onwards. In the incipient etching the chemical action by the salt water had just revealed the crystal structure. The rods were, to begin

with, smoothly polished, so that the action was precisely similar to the etching of a specimen by acid for examination for microstructure. In the deeper etching, however, the material between the grains, and sometimes portions of the grains themselves appeared to have been eroded away to a greater or less degree in different parts. Once the surface of the metal had been attacked the appearance remained the same, however deeply the etching penetrated, and the difference in degree was then only detected by measurement.

The general conclusion was that the degree of etching of these rods depended chiefly upon the duration of the experiment and the velocity of the water, and was not seriously influenced by liberation of gas, by stress, or by annealing; also that the effect observed did not exactly resemble either in character or in intensity the observed action on a propeller blade. Such action as was found appeared not to be attributable in any substantial degree to the cavitation, so that if, as was generally suspected, the erosion of propellers is associated with cavitation, some other and more intense form of cavitation must be sought which was not present to any great extent in these experiments.

(5) WATER HAMMER PRODUCED BY THE CLOSING UP OF VORTEX CAVITIES.

Cases of rapid and deep erosion of propeller blades and the denting of the bosses, notably on the after propellers of the *Cobra* in 1902, and on the after propellers of fast cruisers, as well as instances of undoubted deformation of metal by water hammer in water syrens, air pumps and water pumps, led the committee to desire a theoretical and practical investigation as to the possible pressures that might be produced by the closing up of vortex cavities. The calculations which were made by Mr. S. S. Cook showed that very large pressures might be generated when vortex cavities are caused suddenly to collapse by the final concentration of the initial energy upon a small volume of the fluid. The supposition is made that a certain portion of the fluid occupying, say, a sphere of a known radius is suddenly annihilated, or, what is the same thing, a rigid inner boundary to the fluid enclosing a vacuous space is suddenly removed, and the problem is to obtain the subsequent velocity with which the adjacent fluid flows into the cavity and the instantaneous pressure thus produced.

The calculations show that when a spherical vortex cavity in the sea closes upon a central nucleus of one-twentieth of the diameter of the original cavity, the instantaneous pressure or blow upon this nucleus may reach 68.2 tons per square inch, and if the diameter of the nucleus is one-hundredth of the diameter of the original cavity the pressure may reach 765 tons per square inch. It also is shown that the pressure on impact is independent of the size of the cavities, and is only a function of the ratio of the initial to the final radii, so that very small cavities may cause erosion.*

A review of the instances of propeller erosion shows them not to be inconsistent with such a theory.

In the case of the erosion on the blades of the after propellers of the *Cobra*, there was also intense water hammer on the bosses, the bronze cover plates over the set bolts attaching the propellers to the shaft being driven in with such force as to take an impression of the numbers stamped on the bolt heads. More recently, similar action was observed on the after propellers of some fast cruisers, on those parts of the propeller blades which passed through the wake of the forward propellers. The

* Mr. Cook's calculations and conclusions were confirmed in a paper to the Royal Society by Lord Rayleigh August, 1917.

forward propellers of one of these cruisers which showed no signs of erosion soon began to erode when transferred to the inner shafts, indicating that the erosion was due to the position of the propeller.

An exhaustive macroscopical and microscopical examination was made by Professor Carpenter of the blade of a cruiser's propeller which had been found to be eroded over nearly the entire surface, and which was excessively eroded in three well-defined areas. The effects of water hammer action were looked for, but very little evidence could be obtained owing to the similarity of structure of slightly eroded metal, and that subjected to hammer blows. In two places, however, evidence of somewhat excessive deformation was found which may be attributed to this cause.

An analysis made of the surface layers of the eroded portions showed no difference in composition between them and the body of the underlying metal, and therefore gave no evidence of selective corrosion. This theory of the action seemed also to be borne out by the variable positions of the patches of erosion, which were found on the blade tips, near the center of surface, at the roots, or on the boss; but always in the same places on the blades of the same propeller. The blades in their rotation apparently cut the vortices from the forward propellers, or from the shaft struts, at the same place in turn, which would account for the similarity of pattern observed. When propeller blades are found to be bent, the eating away of the surface is usually found on the side convex to the bend, and the bending might be accounted for by the water-hammer action stretching the surface on that side. Erosion is also found where there is any inequality of the blade surface, *e.g.*, at the trailing edges of shackle holes through blades, and even center-punch marks show the same effect, appearing like little comets, of which the center-punch mark forms the head.

The time element in cases of propeller deterioration is of interest. In one case, at least, the propellers had to be renewed after a three hours' trial. Apparently the action, if it is going to occur, is most severe during the early life of the propeller and proceeds more slowly afterwards. This might be accounted for by the erosion carrying the surface beyond the region of the nuclei of pressure—as in the siren experiment, described later. Rapid erosion has occurred on the casings of air pumps of the drum or Roots blower type, on the delivery side, where the vacuous spaces would close up by the ingress of the sealing water. Also in a screw pump for circulating fresh water through a condenser, where cavitation was known to exist from the usual crackling or water-hammer noises, erosion was found all round the liner and opposite to the blade tips, where cavities caused by the blades would be closed up rapidly.

Also in cases where the blades and hubs of propellers have been working in the wake of forward propellers, they have not only been rapidly eroded, but have in some cases shown the appearance of having been violently hammered, and near the tips the blades have been bodily bent.

To connect the ideal problem with the actual it remains to form a conception of the manner in which the cavities collapse in the actual case. It would seem that for our conclusions to apply we must imagine a discontinuous action, that is, that the vacuous spaces are being continually formed and collapsed, rather than that they disperse along the surface as they move away from the place of their origin. And this may be the explanation of the capriciousness of the phenomenon. In a case, for instance, such as that of the *Cobra's* propellers, where there is reason to suppose that the after propeller blades cut across vacuous spirals trailing from the forward ones, we have here the very conditions for such a discontinuous action, and the pressure on the working faces of the after propeller blades would materially assist the collapse.

CONCLUSIONS.

The conclusions arrived at are that the corrosion of propellers is very slight, but that erosion is serious and is caused by the hammer action of the water on the propeller blades, produced by cavities closing up on the surface of the blades. This action may be caused either by cavitation of the propeller itself, occurring more generally when the propeller is in a varying wake, or by the cavities and vortices formed by the action of other propellers ahead of it, and the erosive action is generally aggravated upon a propeller which works in the wake of another.

The water-hammer action is also likely to be produced when violent and abrupt eddies are formed in the water by the form of stern frame, shaft bossing or "A" brackets, or when the lines are very full and such as to cause an eddying wake. Cavitation will only produce erosion when accompanied by conditions which cause the cavities to collapse in such a way and in such a position that the energy of collapse is concentrated on a small portion of the propeller surface.

From the calculations it appears that the pressure of the water hammer is independent of the size of the cavity, depending only upon the ratio of its contraction, so that the cavities causing the erosion may be large or small. In the case of the propeller of a fast cruiser, inspected by the committee at the works of Messrs. Stone, some of the cavities which caused the erosion must have been large, as the dents observed in places were similar to those produced by using a round-nosed hammer. In the case of a certain single-screw ship of 11 knots speed, whose stern lines are full, the erosion of her propeller may have been caused by the cavities formed by the blades on their passage through the following wake, such cavities being carried with the blades and closed up suddenly when the blades entered those stream-lines of the vessel where the ship was normal, viz: about 11 per cent.

In the case of twin-screw vessels, the propeller should be placed as far as possible from the after end of the bossing-out, consistently with satisfactory mechanical conditions of the support.

In the case of four-shaft vessels, the after propellers should, in addition, be placed as far as possible abaft the leading propellers, and as far as possible clear of the race of the leading propellers.

The form of the after edges of struts and bossings-out of very fast vessels should be designed and placed in reference to the stream-lines of the vessel with great care, so as to cause a minimum of turbulent wake.—"The Engineer."

ELECTRIC DRIVE ON MERCHANT SHIPS.

COMPARISON OF ELECTRIC DRIVE WITH REDUCTION GEARING—ELECTRIC PROPULSION OF SINGLE-SCREW CARGO VESSEL PROPOSED.

By W. L. R. Emmet.*

Justification for electric drive on merchant ships is that it affords a transmission efficiency practically equal to that obtainable with gears of suitable speed reduction, and that it accomplishes this result by a simpler and more reliable means. The writer has given much thought to this problem, both with respect to geared equipments and to electric drive, has conducted many experiments, and obtained much practical experience. A year or two ago he was of the opinion that electric drive would not be commercially justified on low-speed merchant vessels on account of higher cost and a somewhat less transmission efficiency. More correct studies of

* Consulting engineer, General Electric Company, Schenectady, N. Y.

this subject and the development of improved electrical methods have shown that the difference of transmission efficiency is very slight and that the small difference of cost which will apply to a just comparison is much more than justified by many practical advantages accomplished.

Geared turbine applications to merchant ships have been of two kinds: first, single-reduction equipments with relatively low-speed turbines, and, second, double-reduction equipments with high-speed turbines. The General Electric Company was the first to produce equipments of the latter type, which afford a less efficient transmission but admit the use of a very much better and simpler turbine. By the introduction of single-reduction turbine equipments, Parsons showed by comparative tests a reduction in steam consumption of about 19 per cent as compared with the reciprocating engine. Four years ago I predicted a further reduction of 14 per cent by the use of high-speed turbines with double-reduction gearing, and very complete data and tests have proved the correctness of this prediction.

SINGLE-REDUCTION GEARS ABANDONED IN BRITISH SHIPS.

An interesting fact in this connection is that in recent designs Parsons has abandoned the single-reduction method, and the new standard English ships are being built with double-reduction gearing and high-speed turbines. At the same time that this change is being carried on on a large scale in England, the DeLaval Company and others in this country, with the approval of the Emergency Fleet Corporation, have made the opposite change and have abandoned double reduction in favor of single reduction, on the ground that the gearing could be made more reliable. An interesting fact in this connection is that the principal troubles with marine gears have developed in the low-speed element.

These facts illustrate the uncertainties of the gear situation and the unformed state of opinion relating to it.

Machinery which drives a propeller may be subject to very severe shocks and periodic vibrations, and gear troubles have developed on ships in a manner which seems erratic and very difficult of classification.

The efficiency of ship gears has also been overestimated. Careful tests have indicated that the efficiency of transmission in a 2,500-horsepower double-reduction equipment, running smoothly and in good condition, does not exceed 94.5 per cent, and it must be remembered that the equipment is further handicapped by the presence of a reversing turbine which occasions about eight times as much friction resistance as a similar turbine moving in the ahead direction. With high-speed turbines and the most improved electrical apparatus, we can get a transmission efficiency of 93.5 per cent on such a ship, so that it can be safely predicted that in any vessel in which high-speed turbines are used, the difference of transmission efficiency will not exceed one per cent. If, on account of gearing conditions, the turbine has to be divided into two or more parts, there is a further disadvantage through increased packing losses and other complications.

EFFECTS OF TEMPERATURE WITH USE OF REVERSING TURBINES.

Another matter which is important in the comparison of geared and electric ship equipments is the effects of temperature which may arise in connection with the use of reversing turbines, particularly where high degrees of superheat are applied. In an electrically-driven ship the turbine always moves in the same direction and never requires to be stopped except when operation is entirely discontinued. Thus its service is equivalent to that of a turbine on shore. When a turbine is operated in the

reverse direction the friction is at least eight times as great as in the normal direction. The reversing elements of our 2,500-horsepower ship turbines, although very small in diameter and of a minimum rotation loss, will, when operated in a 10-inch vacuum at full speed, heat up enough to turn blue. In our tests we regularly avoid this by introducing water. The introduction of water or any fluid which may limit this heat will naturally cause a large increase of loss.

An experiment which suggests the possibilities of such heat troubles as have been mentioned was recently made in Schenectady. A turbine designed to operate at 3,600 revolutions per minute was forced to revolve in the reverse direction at 2,000 revolutions per minute. A small amount of steam, passed through the buckets in the normal direction while it was being reversed, produced in a short time a temperature of 940 degrees F. in the nozzles of the last stage.

Temperature variations in actual practice in ship turbines are, to a great extent, limited by the heat-storage capacity of the turbine parts themselves, but there are bound to be local heatings and irregular distributions of heat, which, with high superheat, cannot fail to be a source of increased danger. Superheat affords means of greatly increased economy wherever turbines are used, and this advantage should not be sacrificed in ships where fuel economy is of such great importance. In an electrically-driven ship superheat can be a source of no disadvantage to a turbine—in fact, it is likely to increase the life of the blading.

ELECTRICAL PROPULSION LIMITED BY PROPELLER SPEED.

The electrical propulsion of ships may be limited under some conditions by the matter of propeller speed, since we cannot have less than two poles in the generator, and since it is undesirable to have to provide more than 60 or 70 poles in a motor. For ships requiring 3,000 horsepower or over and a speed of about 11 knots, a turbine operating at 3,000 revolutions per minute can be advantageously used, which, with a 60-pole motor, will give 100 revolutions per minute to the propeller, and a high propeller efficiency. In higher-speed vessels the conditions are generally easier. With smaller powers and propellers operating under 100 revolutions per minute it will generally be desirable to use gears, even if electrical transmission is adopted. In that case it is desirable to place the gears between the turbine and the generator instead of between the motors and propellers, as has been done.

DIFFICULTIES IN LOW-SPEED ELEMENTS OF GEARING.

Experience with ship gearing indicates that the greatest difficulties arise in the low-speed elements. This is probably due to the fact that these elements receive more direct shocks from the propeller, and also that they give a much less perfect lubrication effect. There is presumably an effective oil cushioning in high-speed gears which is absent in low-speed gears.

In electric drive equipments it would generally be desirable to furnish direct current exciting current from a separate source, and it has been thought desirable to combine excitation, lighting and the driving of auxiliaries. The auxiliaries on most existing ships involve a considerable unnecessary loss of power, and by driving them electrically and exhausting a well-designed auxiliary generating unit into the feed heater and into an intermediate stage of the turbine, a very good efficiency and simple operating condition can be afforded.

The following copy of a proposal recently made illustrates the possibilities of an application to a 4,000-horsepower vessel with a single screw operating at 110 revolutions per minute:

PROPOSAL FOR ELECTRIC DRIVE ON 4,000-HORSEPOWER MERCHANT VESSEL.

"The auxiliary generating units are of such a size that one of them will be capable of affording lighting and excitation and all the auxiliary power which has been contemplated while the ship is at sea. These units are designed to run condensing or non-condensing with a good water rate. By carrying as a spare part a suitable direct-current motor and gear which could be connected to the end of the main shaft, these auxiliary units are used to propel the ship at nearly half speed in case the main unit or main motor were unfit for service. It would in any case be desirable to have auxiliary condensing facilities for use in port, and these could be used in such an emergency.

"The main propulsion equipment consists of:

	Approximate Net Weight, Pounds
One 3,200-k. v. a., 3,500 revolutions per minute, 2,500-volt turbine generating unit	82,000
One 4,000-horsepower, 110 revolutions per minute, alternating-current motor	80,000
Instruments and controlling mechanism for the above	9,000

"Our calculations concerning the performance of this equipment show the following results, and, since the efficiency of every part is thoroughly understood, it is certain that these results can be actually obtained in practice. When the turbine is operating with a steam pressure of 265 pounds gage at the throttle, 150 degrees F. superheat, and 28.5 inches vacuum, we should obtain a brake horsepower delivered to the propeller shaft with 9.4 pounds of steam, not including auxiliary steam.

"In giving guarantees, we must make an allowance for possible errors in testing and will guarantee that this water rate, under these conditions, will not exceed 10 pounds per shaft horsepower hour.

Auxiliaries.—We have assumed that the boiler-feed pump is steam-driven and requires 2,500 pounds of steam per hour, also that a Blake twin-beam wet and dry-air pump is used which consumes 600 pounds of steam, making a total of 3,100 pounds of steam from these pumps for feed-water heating. The following electrically-driven auxiliaries have been assumed:

	Motor speed.	Horsepower.	Electrical Input, Horsepower.
Circulating pump	1,700/1,000	75.00	82.5
Excitation		80.00	80.0
Lighting		20.00	20.0
Oil Cooler	1,700	1.25	1.5
Fire room blowers	850	20.00	22.5
Steering	800	5.35	6.0
Continuous load			212.5 = 157.0 kw.

	Motor speed.	Maximum Horsepower.	Average Continuous Horsepower.
Sanitary pump	1,700	5.0	2.5
Fresh water pump	1,700	5.0	2.5
Refrigerating pump	1,700	5.0	2.5

Evaporator pump	1,700	1.0	1.0
Workshop	1,700	5.0	2.5
Bilge pump	850	5.0	5.0
			16.0
Total input		18.0 =	13.5 kw.
Total maximum load			170.5 kw.

"In addition to the above motors there is included one 20-horsepower, 900 revolutions per minute induction motor for operating the blower for main motor ventilation. In connection with the above motors, while we are including hand-operated starting devices, no mechanical parts such as pumps, blowers, etc., are included. In other words, only motors and starters are included.

"We propose to drive these auxiliaries from two 150-kilowatt, direct-current sets, operating non-condensing against a back pressure of 5 pounds gage, with part of this exhaust used to bring up the feed-water temperature to 212 degrees F. and the remainder going into a suitable stage in our turbine for additional work.

"The total flow of steam from the boiler at maximum speed will be 43,000 pounds per hour. Assuming 14 pounds of steam evaporated per pound of oil (same efficiency as Babcock & Wilcox boilers on battleship *Wyoming*), this would require 3,100 pounds of fuel oil per hour, or 33.25 tons (2,240 pounds) for 24 hours. Thus the water rate, including all auxiliaries, lighting, and steam required for heating feed water, should be 11.1 pounds per shaft horsepower hour.

"The electrical apparatus proposed for this purpose is of an extremely simple and reliable type. Our records over long periods of years covering such apparatus show that only about one-tenth of one per cent of the motors and generators of such voltage, which are installed in all kinds of service, show any electrical trouble in a period of ten years.

"The turbine is of the most efficient type manufactured for such a capacity and is designed with very liberal factors of safety with all the most recent features of construction. Such a turbine is capable of running at a considerable proportion of its capacity if one of its many wheels is in operating condition. Packings, bearing sleeves and other perishable parts are easily replaceable, so that under almost any combination of circumstances such a turbine can be quickly got into operative condition after it has been accidentally damaged.

"The standards of reliability are so high with such apparatus that it is believed that they occasion less risk of stoppage than is incurred in single-screw ships propelled in any other manner. In case, however, the novelty of such an equipment raises doubts which may increase insurance rates, a motor can be provided as stated above which will propel the ship at reduced speeds from power delivered by the auxiliary generating units.

"It is believed that the operation of these relatively large auxiliary generating units and the use of motors for driving the auxiliaries mentioned are justified from the standpoint of economy. Generating units of such a capacity will give a very good efficiency, and their operation involves no more complication of detail than is occasioned by the use of the smallest steam turbine—in fact, the details of the larger turbine are simpler and better than those of the smaller one."

The proposal quoted above describes an equipment which is in itself much less liable to trouble or interruption than any type of existing equipment used on a single-screw ship, and at the same time affords, at very

small cost for spares, means by which the ship can be run if the main machinery is inoperative. The figures given as to results are correct and dependable, and are so good that their accomplishment in many classes of service would pay for the change in three years. With very few exceptions, the prevailing opinion of the most experienced marine engineers is that the reciprocating engine for ship propulsion is obsolete. Its replacement is inevitable, and the General Electric Company, with its experience and facilities for development, is ready to do the work.—"Marine Engineering."

OBITUARY.

CHIEF ENGINEER CHARLES H. MANNING, U. S. N.

Chief Engineer Charles H. Manning died at his home, in Manchester, N. H., on the 8th of April, 1919.

He was born in Maryland in 1842, where his people had lived for more than a century. He was graduated at Harvard University, in the class with John D. Long (afterwards Secretary of the Navy), and Eugene Hale (so long a U. S. Senator).

He entered the Navy as a Third Assistant Engineer in February, 1863, and was assigned to special duty in Baltimore. It was an unfortunate order for him, as it delayed his promotion, having deprived him of the essential sea service. In 1865 he was assigned to the *Dacotah*, on which vessel he served three years, a period necessary for his promotion to Second Assistant Engineer. His next cruise was on board the *Seminole*, on the Home Squadron. He was promoted to Passed Assistant Engineer in 1872, and ordered to the Naval Academy as instructor.

In 1875 he was ordered to the *Swatara*, the first ship of the Navy to make a voyage with a compound engine. The voyage was to the Kerguelen Islands, in the South Sea, for the purpose of the observation of the transit of Venus, and was a hazardous voyage, but productive of grand results.

His next duty was at the Naval Academy, where he remained three years as instructor. He served as Chief Engineer on board the *Dispatch* 1881 and 1882, after which he was granted leave of absence, which was extended from time to time. During this period he was under treatment for deafness. The Navy Department favored Chief Engineer Manning, as his services were so valuable and it was desired to retain him on the active list.

He was retired in June, 1884, and at once was made Chief Engineer of the famous Amoskeag Manufacturing Establishment at Manchester, at a salary more than three times what he had received on the active list of the Navy. He was promoted to the rank of Chief Engineer in the Navy (on the retired list), with the rank of Lieutenant Commander, by act of Congress approved March 4, 1911.

The many improvements he made at the Amoskeag works, his success as a hydraulic engineer and his willingness to lend a helping hand to all his friends, soon brought him in the limelight as a Consulting Engineer.

Mr. Manning retired from business about four years ago, having become so deaf that he felt it unjust to his employers to continue. Personally he was one of the most popular men in his corps and one of the brightest and best informed. He was a member of this Society since its organization. He was a member of the Society of Mechanical Engineers, Naval Architects and Marine Engineers, and took part in their debates.—G. W. B.

MARSHALL T. DAVIDSON.

The death of Marshall Ten-Broeck Davidson occurred at his residence in Brooklyn on the 10th of April, after a long illness. He was born in Albany in 1837, but his early life was spent in Hudson. His paternal ancestry was Scotch, dating from the colonial period, and his maternal ancestry was Holland-Dutch, dating from the time of the Dutch occupation of New Amsterdam, afterwards New York City. His intimate acquaintance with the Engineers of the Navy and his interest in the profession of Steam Engineering began in his boyhood.

He was educated in the public schools of Hudson and in the Hudson Academy, and he learned the trade of machinist in New York City. On reaching his majority he went to California as an Engineer on board the steam ship *Hudson*. He

remained in this service, which traded in Puget Sound, the Columbia River and Bay of San Francisco, until the Civil War broke out, when he was employed to superintend the building of two army transports, the *Arizona* and *Clinton,* which had been taken over from the Harris and Morgan Company. When the *Arizona* was completed he became her Chief Engineer, and carried troops to the Gulf of Mexico for General Banks' Expedition to the Teche Country, where the writer first met him. He later became a Chief Engineer in the Revenue Cutter Service and later became the President of the Davidson Steam Pump Company in Brooklyn, which position he held until his death.

The Davidson Pump soon became famous, and though its price was greater than its competitors, there was sale for all that could be built. His high-duty engines for city water works, some of a capacity of 40,000,000 gallons a day, were famous.

He married Miss Bame, of Hudson, who died in 1881, leaving two daughters, one of whom (the Countess Seckendorff) survives. There are five grandchildren.

He was a member of The Union League, Hanover, Brooklyn and Manhattan Clubs, the American Society of Navy Engineers, the American Society of Mechanical Engineers, the Society of Naval Architects and Marine Engineers, and The Grand Army of the Republic.

Personally he was charming and magnetic, and had hosts of friends. He was always delighted to do a good and generous act.—G. W. B.

BOOK REVIEW.

"WRINKLES IN PRACTICAL NAVIGATION," by T. S. LECKY, Eighteenth Edition, revised and enlarged by WILLIAM ALLINGHAM, reprinted by arrangement with the English publishers, D. VAN NOSTRAND Co., $5.00 net.

Lecky's "Wrinkles" needs no introduction. First published in 1881, successive editions have been deservedly popular. The method of treatment employed is in sharp contrast to that of the "American Practical Navigator" and Muir's "Navigation and Compass Deviations," but the "wrinkles" and short cuts disclosed make the book worth the purchase price.

In this edition the greater part of the book is about the same as previous editions. One new chapter, "New Meteorological Measures for Old," discusses at some length the proposed adoption of the metric system and centesimal circular and angular units. Four new appendices are added.

It is unfortunate that a book as widely known as Lecky's "Wrinkles" should, in a 1919 edition, fail to enlarge upon the "computed altitude" methods of obtaining a line of position. The Marc St. Hilaire method, using either a sine-cosine or haversine formula or the Aquino tables, has eliminated much of the drudgery of working up sights, with the added advantage of obtaining a *good* line of position for any azimuth of the observed body.—M. W. B.

ASSOCIATION NOTES.

The following members and associates have joined the Society since the publication of the last number of the JOURNAL:

MEMBERS.

Barr, William M., 116 W. 39th St., New York City.
Brown, Ray E., Lieut. (CC), U. S. N.
Fallon, H. N., Lieut., U. S. N.
Holden, C. F., Lieut., U. S. N.
Rasch, F. W., Lieut., U. S. N.

ASSOCIATES.

Boyd, W. L., Lieut. Commander, R. F.
Price, Joseph, 203 West 11th St., New York City.
Robinson, R. L., 209 Addison Avenue, Palo Alto, Calif.
Thomas, George S., Wm. Cramp & Sons Co., Philadelphia, Pa.
Wardrop, G. Douglas, Managing Editor, Aerial Age, 280 Broadway, New York City.

Please do not fail to keep the Secretary-Treasurer advised of your address.

REPORT ON THE AMERICAN SOCIETY OF NAVAL ENGINEERS FOR THE YEAR ENDED DECEMBER 31, 1918.

Date: March 24, 1919.

THE AMERICAN AUDIT COMPANY.
F. W. LAFRENTZ, C. P. A. (N. Y.), *President.*
THEO. COCHEU, JR., C. P. A. (N. Y.), *Vice-President.* A. F. LAFRENTZ, *Sec'y and Treas*
C. R. CRANMER, *Resident Manager.*

TELEPHONE: 2705 MAIN.
CABLE: AMDIT, NEW YORK.
COLORADO BUILDING, WASHINGTON, D. C., March 24, 1919.

THE COUNCIL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS,
Washington, D. C.

Dear Sirs: We have audited the books and accounts of THE AMERICAN SOCIETY OF NAVAL ENGINEERS for the year ended December 31, 1918, and submit our report, including Exhibits, as follows:
Exhibit "A."—Balance Sheet as at December 31, 1918.
"B."—Statement of Income and Expenditure for the year ended December 31, 1918.

The result of the operation of the Society for the year shows a net profit of $4,269.71.

We checked the cash to March 20, 1919, at which date the Cash Book called for $8,056.57, of which $7,858.57 was on deposit with the American Security and Trust Company; the balance, $198.00, represented receipts not deposited.

We verified the securities of the Society by actual inspection.

Our examination of the financial operations of the Society, together with the assurances of the Secretary, shows that the Society was on a cash basis at December 31, 1918, with no liabilities.

Respectfully submitted,

THE AMERICAN AUDIT COMPANY,
By C. R. CRANMER, *Resident Manager.*

Approved:
F. W. LAFRENTZ,
President.

[SEAL.]

Attest:
C. W. GOETCHINS,
Ass't Secretary.

EXHIBIT "A."

AMERICAN SOCIETY OF NAVAL ENGINEERS,

Washington, District of Columbia.

Balance Sheet, As at December 31, 1918.

ASSETS.

Cash in bank	$3,527.09	
Certificate of deposit, Munsey Trust Company	1,000.00	
		$4,527.09
Account Receivable:		
Dues	$1,496.25	
Publication sales	111.24	
Subscriptions	2,062.14	
Advertisements	1,708.00	
		5,377.63
Investments:		
Army and Navy Club bonds	$1,000.00	
Liberty Bonds, First Issue	1,000.00	
Liberty Bonds, Third Issue	1,000.00	
Liberty Bonds, Fourth Issue	500.00	
Washington Railway & Electric Co. bonds	4,320.00	
War Saving Stamps	838.00	
		8,658.00
Furniture and fixtures		50.84
Gold medal		16.00
		$18,629.56

LIABILITIES.

Surplus:		
Balance, January 1, 1918	$14,359.85	
Net profit for the year ended December 31, 1918, Exhibit "B"	4,269.71	
		$18,629.56
		$18,629.56

EXHIBIT "B."

AMERICAN SOCIETY OF NAVAL ENGINEERS,

Washington, D. C.

Statement of Income and Expenditure, for the year ended December 31, 1918.

INCOME.

Dues		$5,846.66
Publication:		
Subscriptions	$4,221.90	
Sales	585.79	
Advertisements	2,363.50	
Exchange	2.89	
		7,174.08
Old dues collected		102.45
Interest on investments		472.16
		$13,595.35

EXPENDITURE.

Publication:		
Printing	$5,445.20	
Engraving	670.43	
Drafting	20.00	
Manuscript	883.50	
Postage and expressage	220.34	
Commissions	156.92	
		$7,396.39
General Expense:		
Salaries	$1,620.00	
Stationery and printing	111.85	
Certificates	17.40	
Rent, safe deposit box	5.00	
		1,754.25
Dues charged off		175.00
Net Profit for year ended December 31, 1918:		
Carried to Exhibit "A"		4,269.71
		$13,595.35